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Ashida et al.

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(45) **Date of Patent: Apr. 20, 2004**

(54) **ENGINE CONTROL APPARATUS**

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(21) Appl. No.: **10/400,397**

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(30) **Foreign Application Priority Data**

Apr. 19, 2002 (JP) 2002-117080

(51) **Int. Cl.**[7] **F02B 31/08**

(52) **U.S. Cl.** **123/432**; 123/491; 123/678; 123/308

(58) **Field of Search** 123/308, 432, 123/491, 678

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(74) *Attorney, Agent, or Firm*—Foley & Lardner

(57) **ABSTRACT**

The invention comprises at least two independent intake ports **14***a*, **14***b* connected to a combustion chamber **15**, two intake valves **12***a*, **12***b*, and two exhaust valves **13***a*, **13***b*. A first fuel injection valve **31***a* is provided in the first intake port **14***a* and a second intake control valve **26** is provided for opening and closing the second intake port **14***b*. When the engine is in the cold condition, the second intake control valve **26** is slightly opened and fuel is injected from the first fuel injection valve **31***b*. Air-fuel mixture from the first intake port **14***a* and a smaller amount of fresh air from the second intake port **14***b* are introduced [into the combustion chamber], and the overall air-fuel ratio inside the combustion chamber is controlled to become slightly lean.

15 Claims, 12 Drawing Sheets

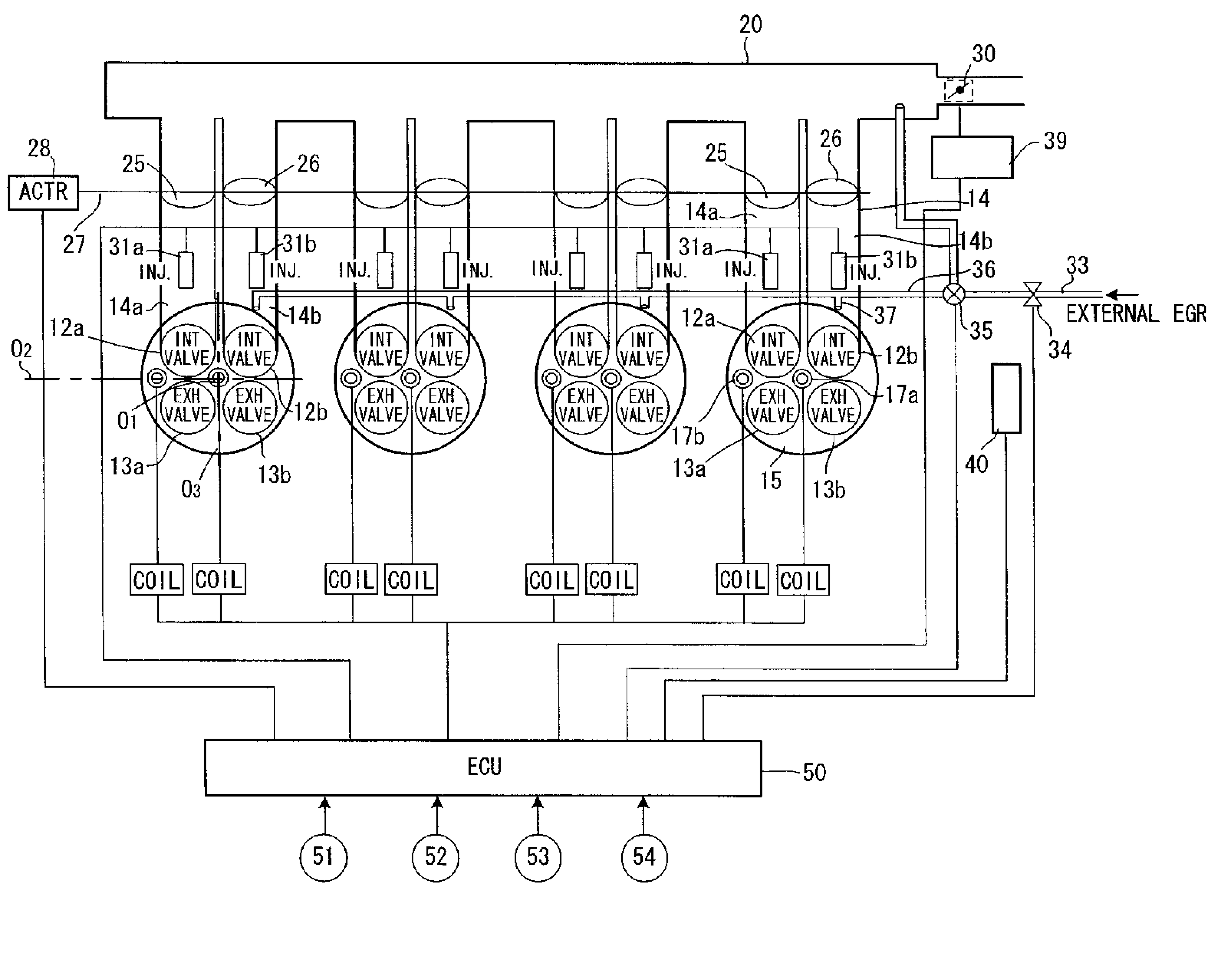


FIG. 1

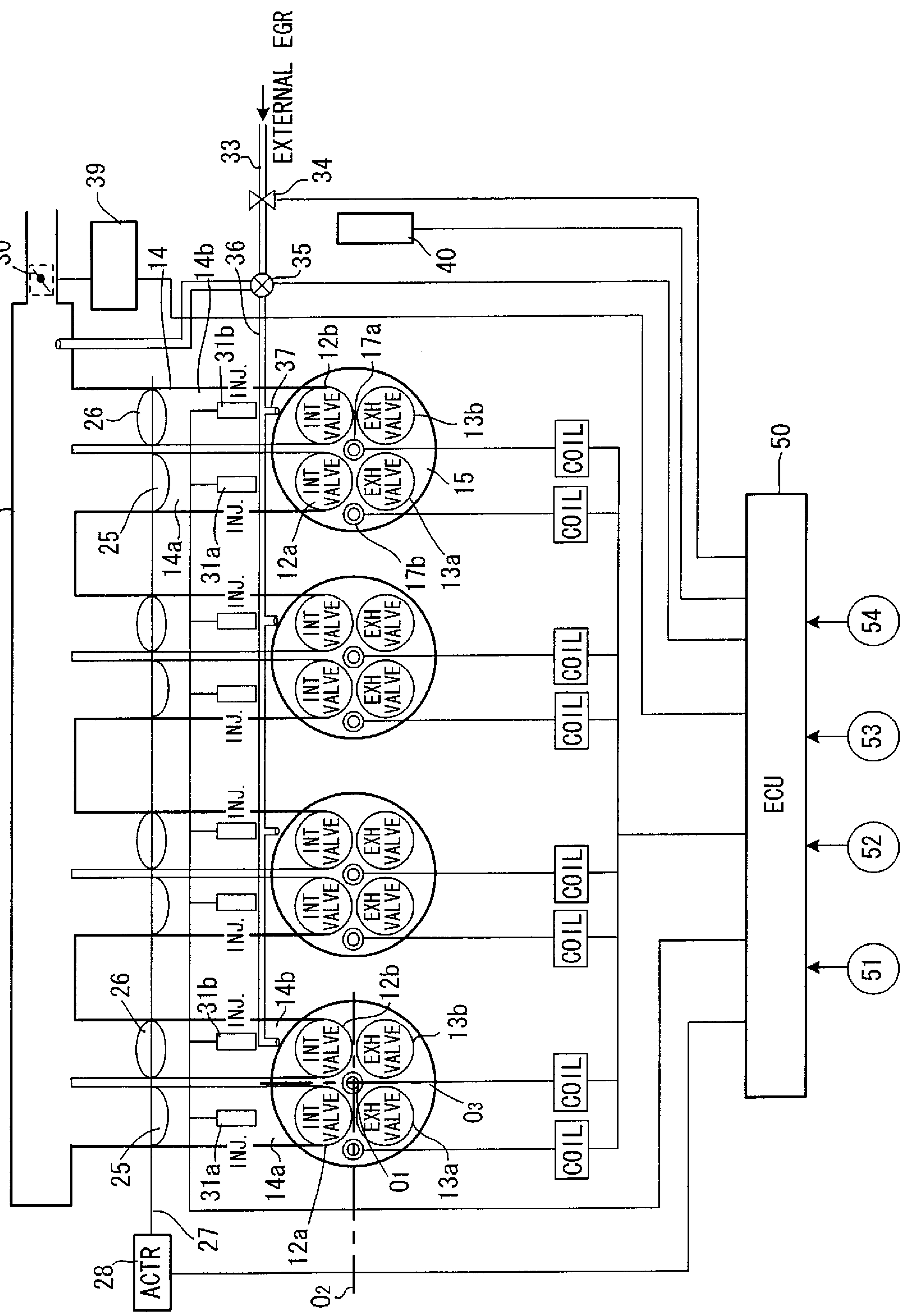

EXTERNAL EGR
33
34
35
36
37
39
40
30
14
14b
14a
20
25
26
27
28
ACTR
31a
31b
INJ.
INT VALVE
EXH VALVE
12a
12b
13a
13b
15
17a
17b
COIL
50
ECU
51
52
53
54
O_1
O_2
O_3

FIG. 2

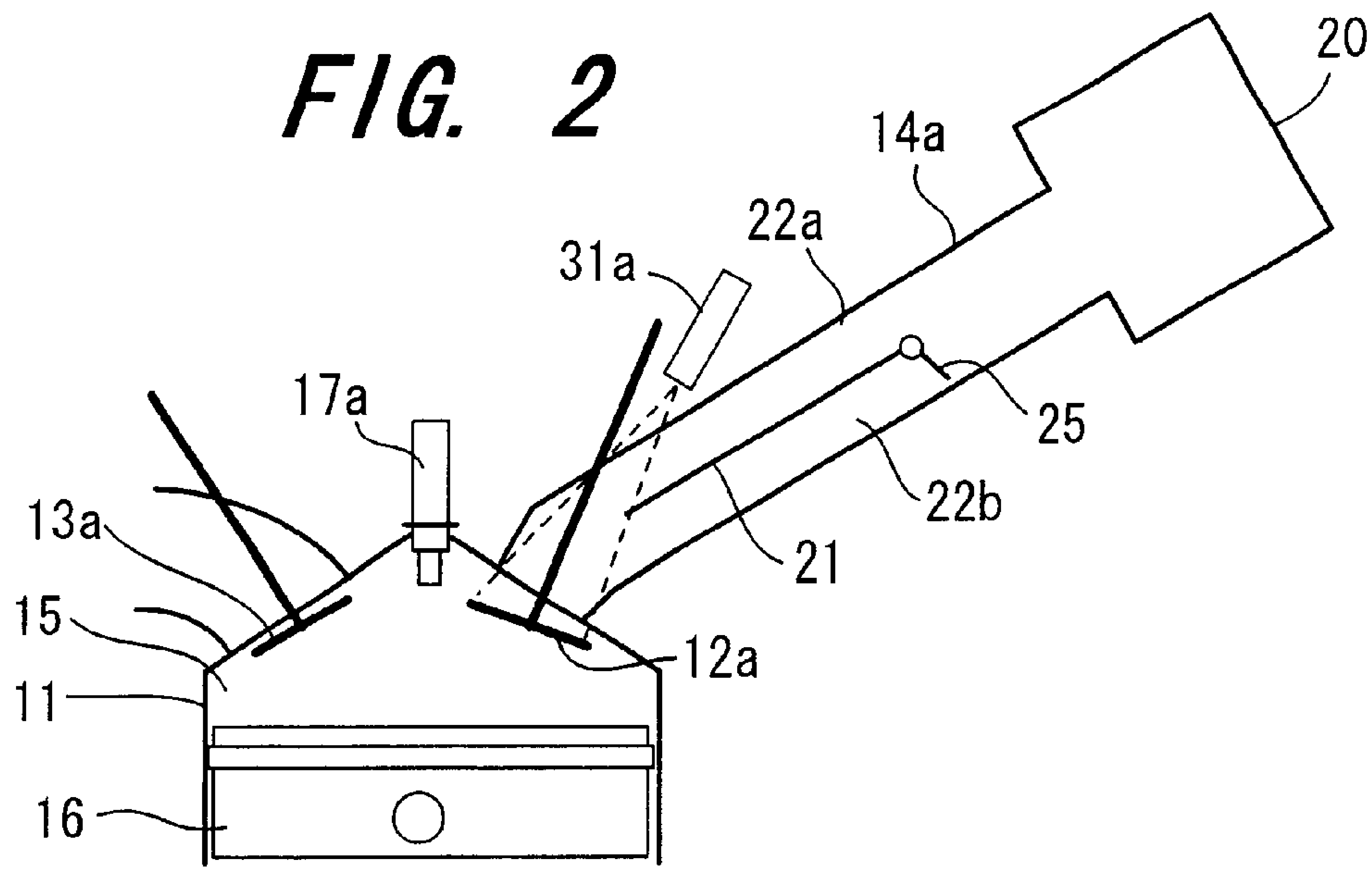

20
14a
22a
31a
17a
25
22b
13a
21
15
12a
11
16

FIG. 3

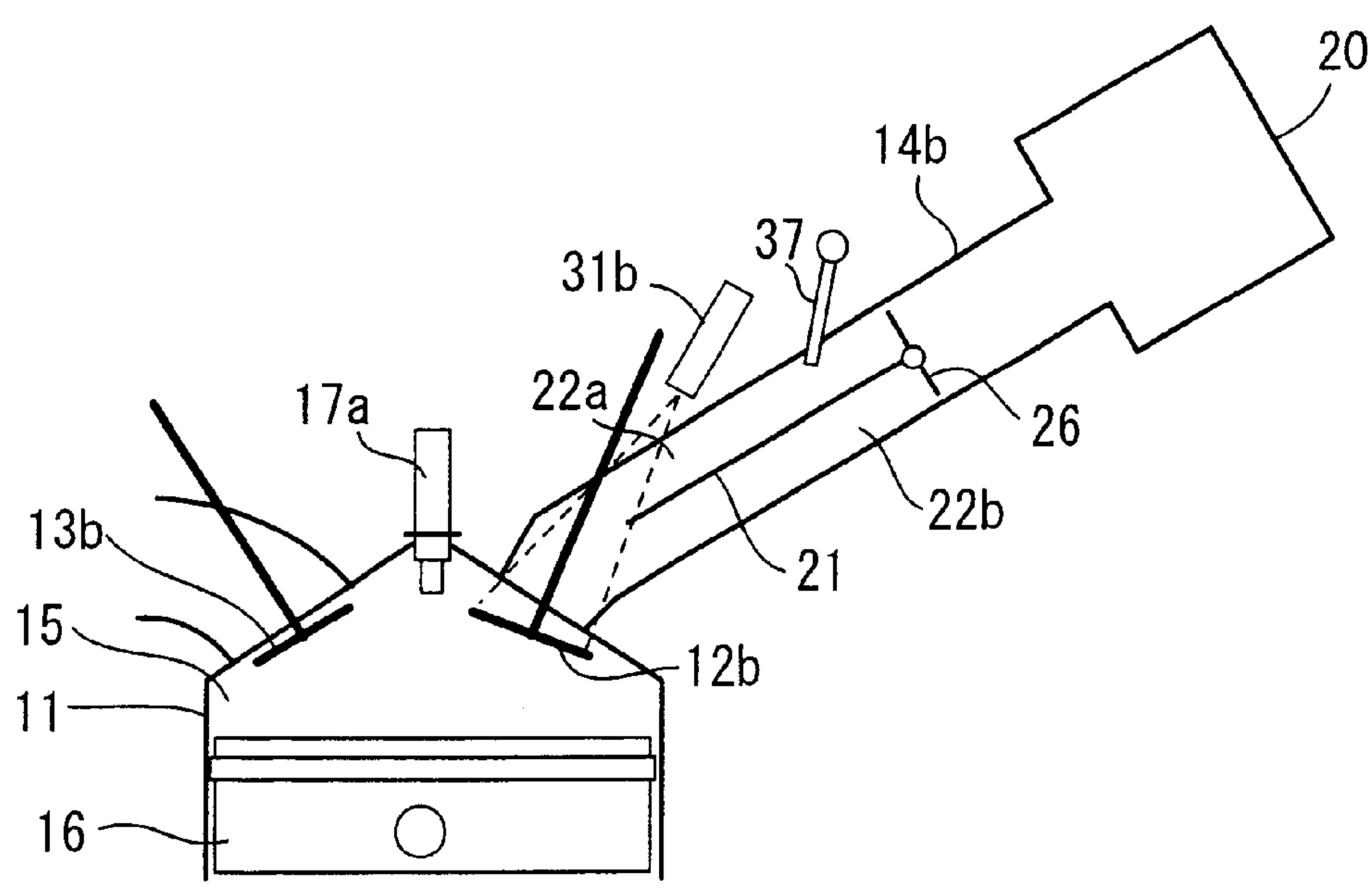

20
14b
37
31b
17a
22a
26
22b
13b
21
15
12b
11
16

FIG. 4

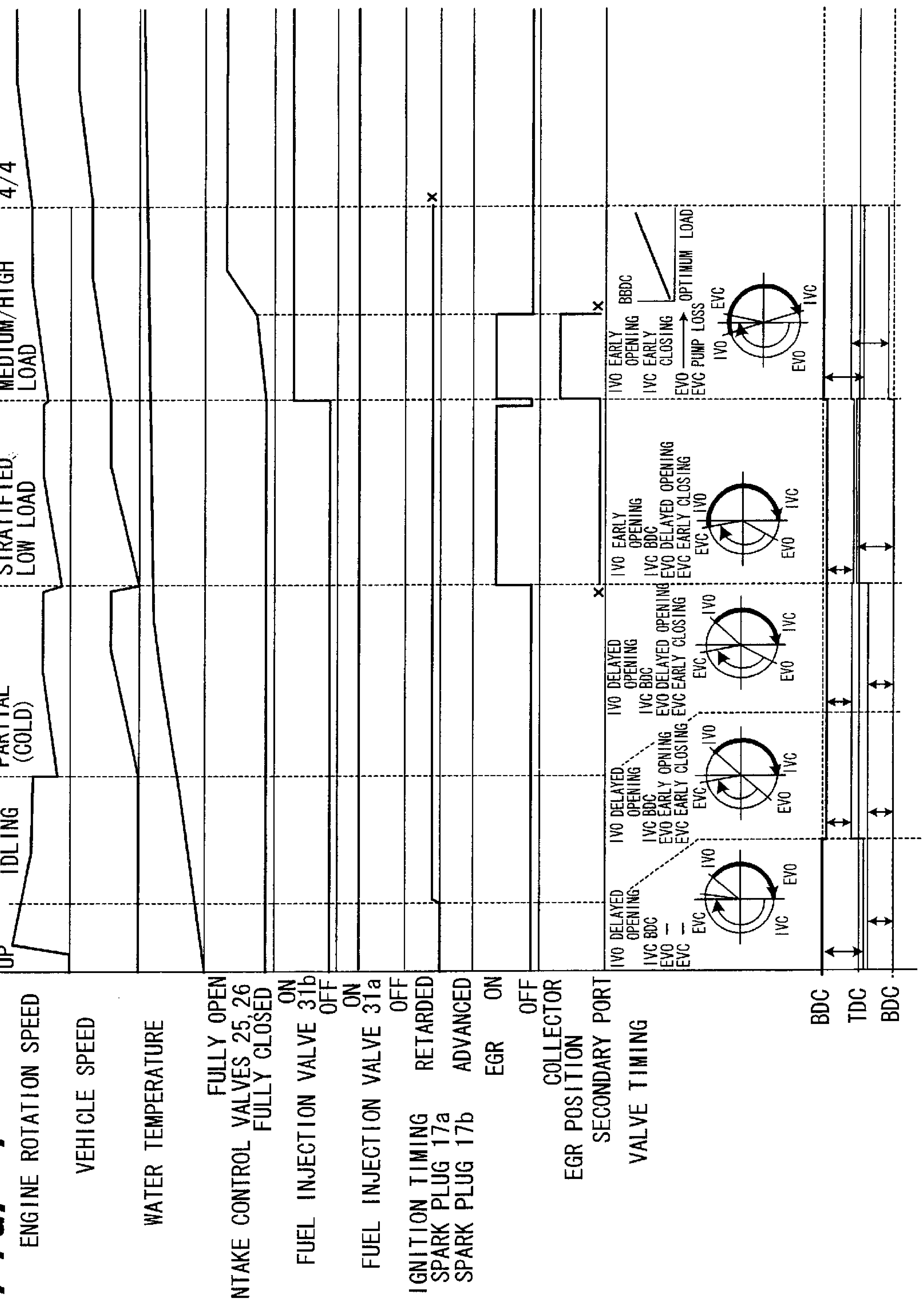

ENGINE ROTATION SPEED
VEHICLE SPEED
WATER TEMPERATURE
FULLY OPEN
INTAKE CONTROL VALVES 25, 26
FULLY CLOSED
ON
FUEL INJECTION VALVE 31b
OFF
ON
FUEL INJECTION VALVE 31a
OFF
IGNITION TIMING
SPARK PLUG 17a
SPARK PLUG 17b
RETARDED
ADVANCED
EGR ON
OFF
COLLECTOR
EGR POSITION
SECONDARY PORT
VALVE TIMING
BDC
TDC
BDC
START-UP
FAST IDLING
PARTIAL (COLD)
PARTIAL (HOT) STRATIFIED LOW LOAD
PARTIAL (HOT) MEDIUM/HIGH LOAD
4/4
IVO DELAYED OPENING
IVC BDC
EVO —
EVC —
IVO DELAYED OPENING
IVC BDC
EVO EARLY OPNING
EVC EARLY CLOSING
IVO DELAYED OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC EARLY CLOSING
EVO → OPTIMUM
EVC PUMP LOSS
BBDC
LOAD
IVO
IVC
EVO
EVC

FIG. 5

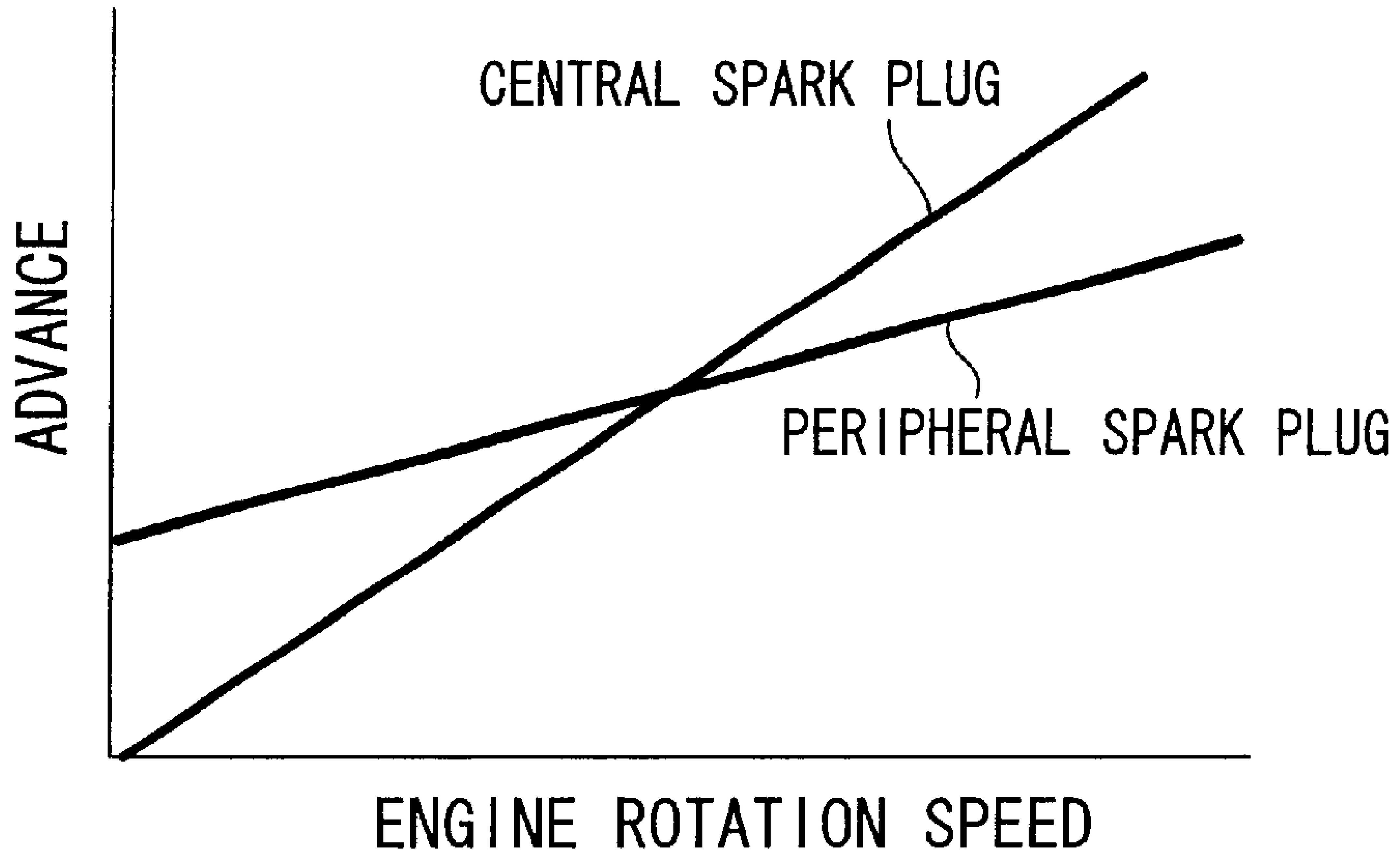

CENTRAL SPARK PLUG
ADVANCE
PERIPHERAL SPARK PLUG
ENGINE ROTATION SPEED

FIG. 6

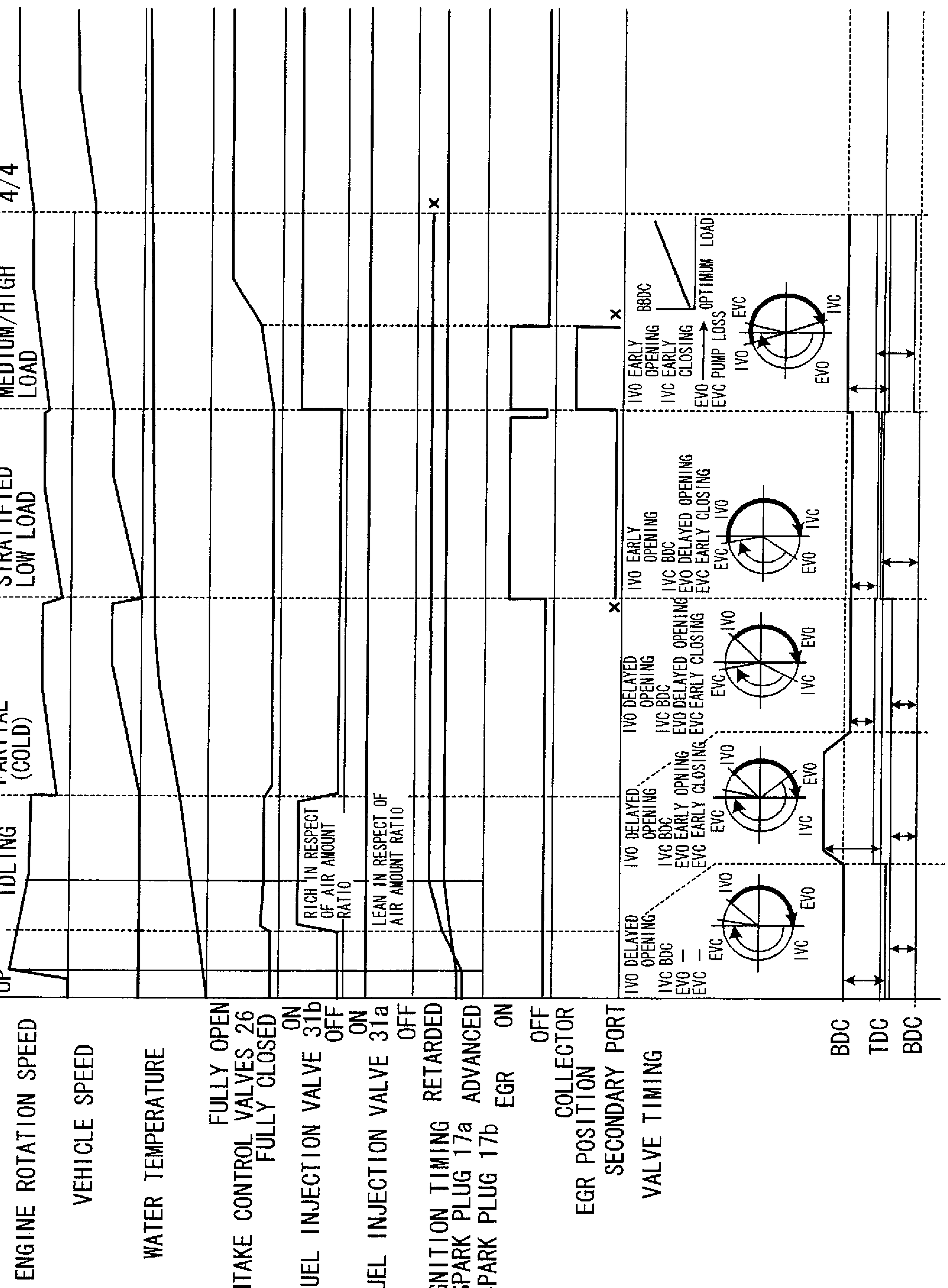
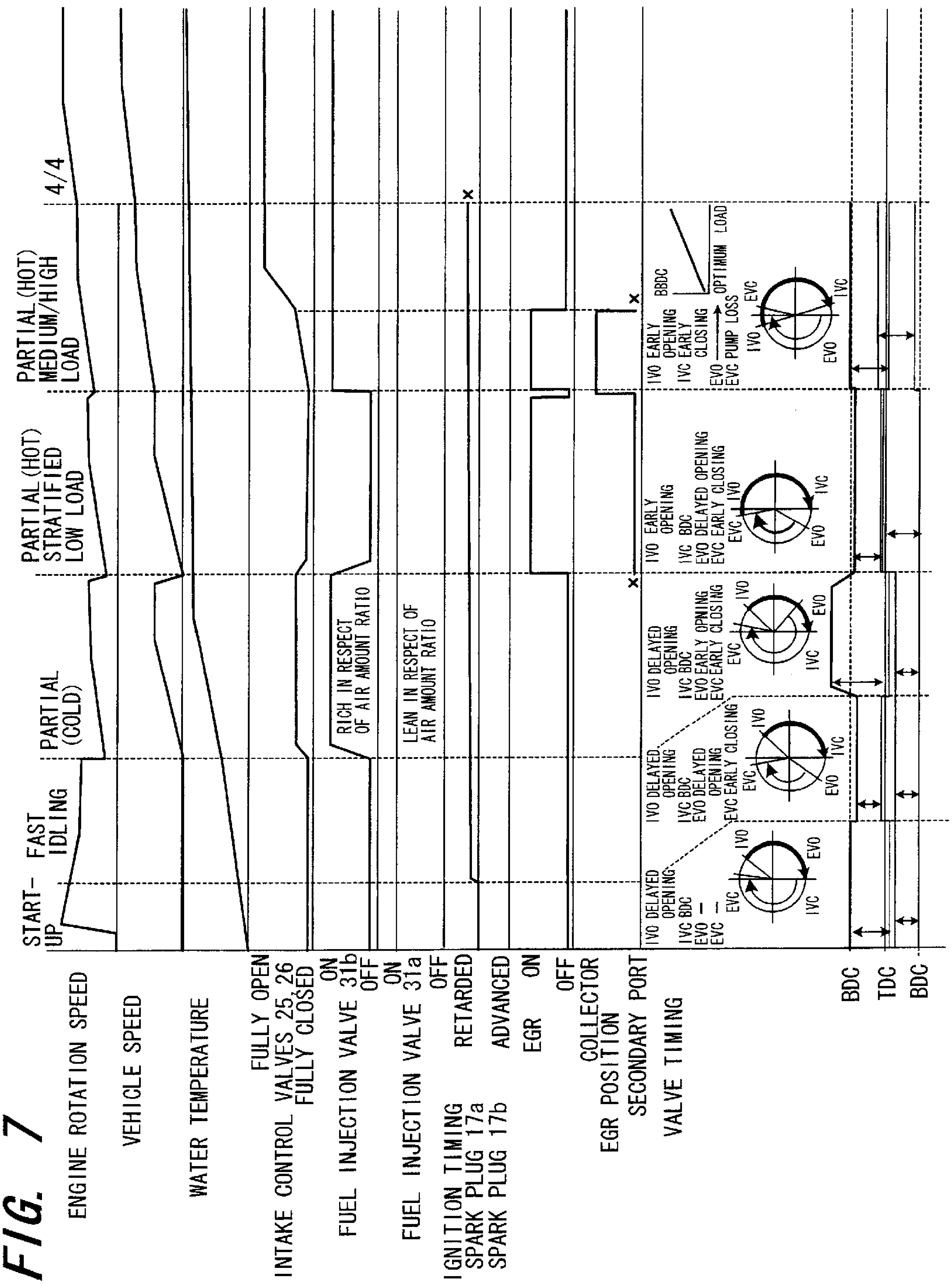

ENGINE ROTATION SPEED
VEHICLE SPEED
WATER TEMPERATURE
FULLY OPEN
INTAKE CONTROL VALVES 26
FULLY CLOSED
ON
FUEL INJECTION VALVE 31b
OFF
ON
FUEL INJECTION VALVE 31a
OFF
IGNITION TIMING RETARDED
SPARK PLUG 17a ADVANCED
SPARK PLUG 17b
EGR ON
OFF
COLLECTOR
EGR POSITION
SECONDARY PORT
VALVE TIMING
BDC
TDC
BDC
START-UP
FAST IDLING
PARTIAL (COLD)
PARTIAL (HOT) STRATIFIED LOW LOAD
PARTIAL (HOT) MEDIUM/HIGH LOAD
4/4
RICH IN RESPECT OF AIR AMOUNT RATIO
LEAN IN RESPECT OF AIR AMOUNT RATIO
IVO DELAYED OPENING
IVC BDC
EVO —
EVC —
IVO DELAYED OPENING
IVC BDC
EVO EARLY OPENING
EVC EARLY CLOSING
IVO DELAYED OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC EARLY CLOSING
EVO
EVC
PUMP LOSS
OPTIMUM LOAD
BBDC
IVO
IVC
EVO
EVC

FIG. 7



ENGINE ROTATION SPEED
VEHICLE SPEED
WATER TEMPERATURE
FULLY OPEN
INTAKE CONTROL VALVES 25, 26
FULLY CLOSED
ON
FUEL INJECTION VALVE 31b
OFF
ON
FUEL INJECTION VALVE 31a
OFF
IGNITION TIMING
SPARK PLUG 17a
SPARK PLUG 17b
RETARDED
ADVANCED
EGR ON
OFF
EGR POSITION
COLLECTOR
SECONDARY PORT
VALVE TIMING
BDC
TDC
BDC
START-UP
FAST IDLING
PARTIAL (COLD)
PARTIAL (HOT) STRATIFIED LOW LOAD
PARTIAL (HOT) MEDIUM/HIGH LOAD
4/4
RICH IN RESPECT OF AIR AMOUNT RATIO
LEAN IN RESPECT OF AIR AMOUNT RATIO
IVO DELAYED OPENING
IVC BDC
EVO —
EVC —
IVO DELAYED OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO DELAYED OPENING
IVC BDC
EVO EARLY OPNING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC BDC
EVO DELAYED OPENING
EVC EARLY CLOSING
IVO EARLY OPENING
IVC EARLY CLOSING
EVO
EVC PUMP LOSS
OPTIMUM LOAD
BBDC
IVO
IVC
EVO
EVC

FIG. 8

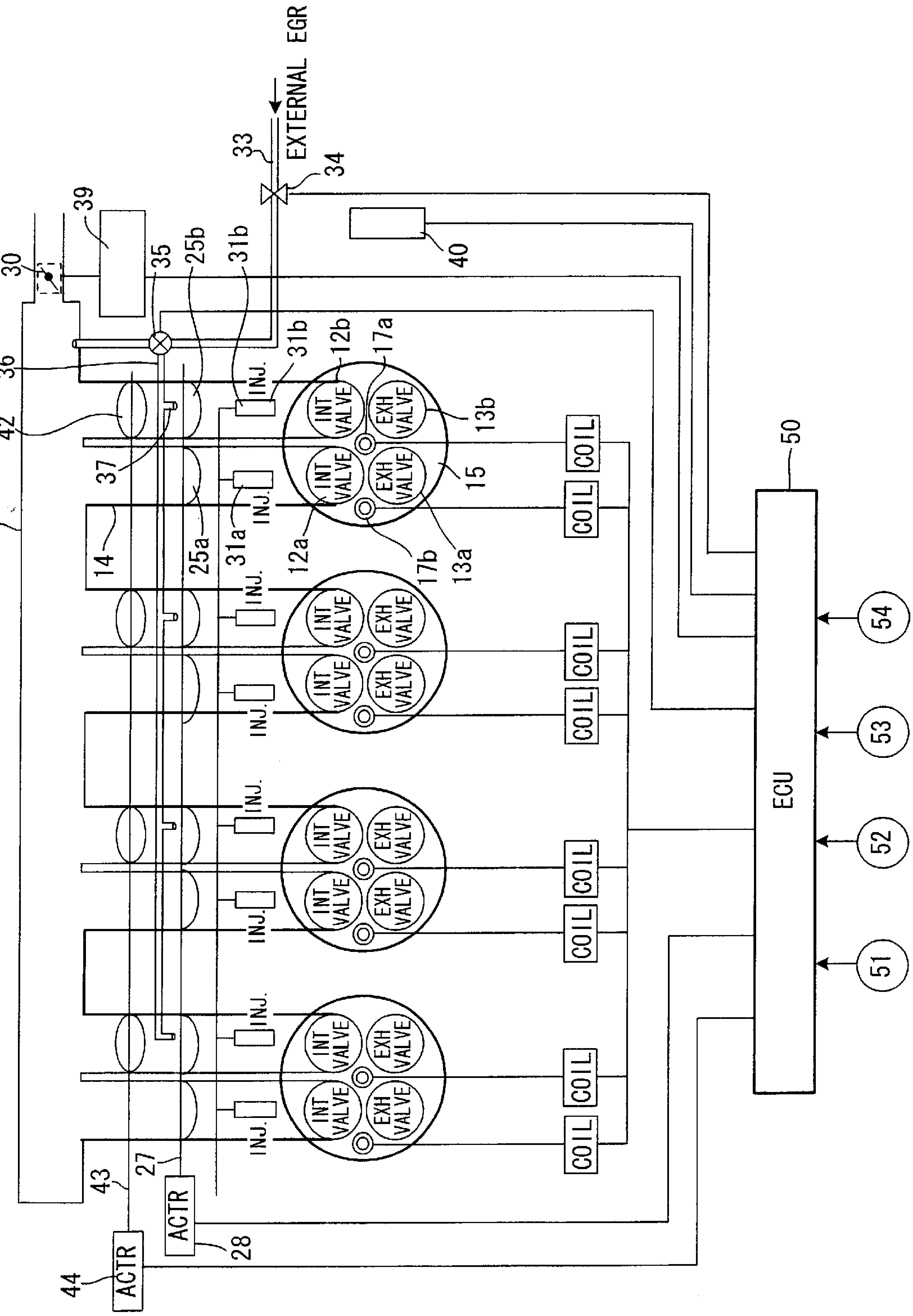

EXTERNAL EGR
33
34
39
30
35
25b
31b
36
42
37
20
14
25a
31a
12a
12b
17a
17b
13a
13b
15
40
50
ECU
51
52
53
54
INJ.
INT VALVE
EXH VALVE
COIL
ACTR
43
27
28
44

FIG. 9

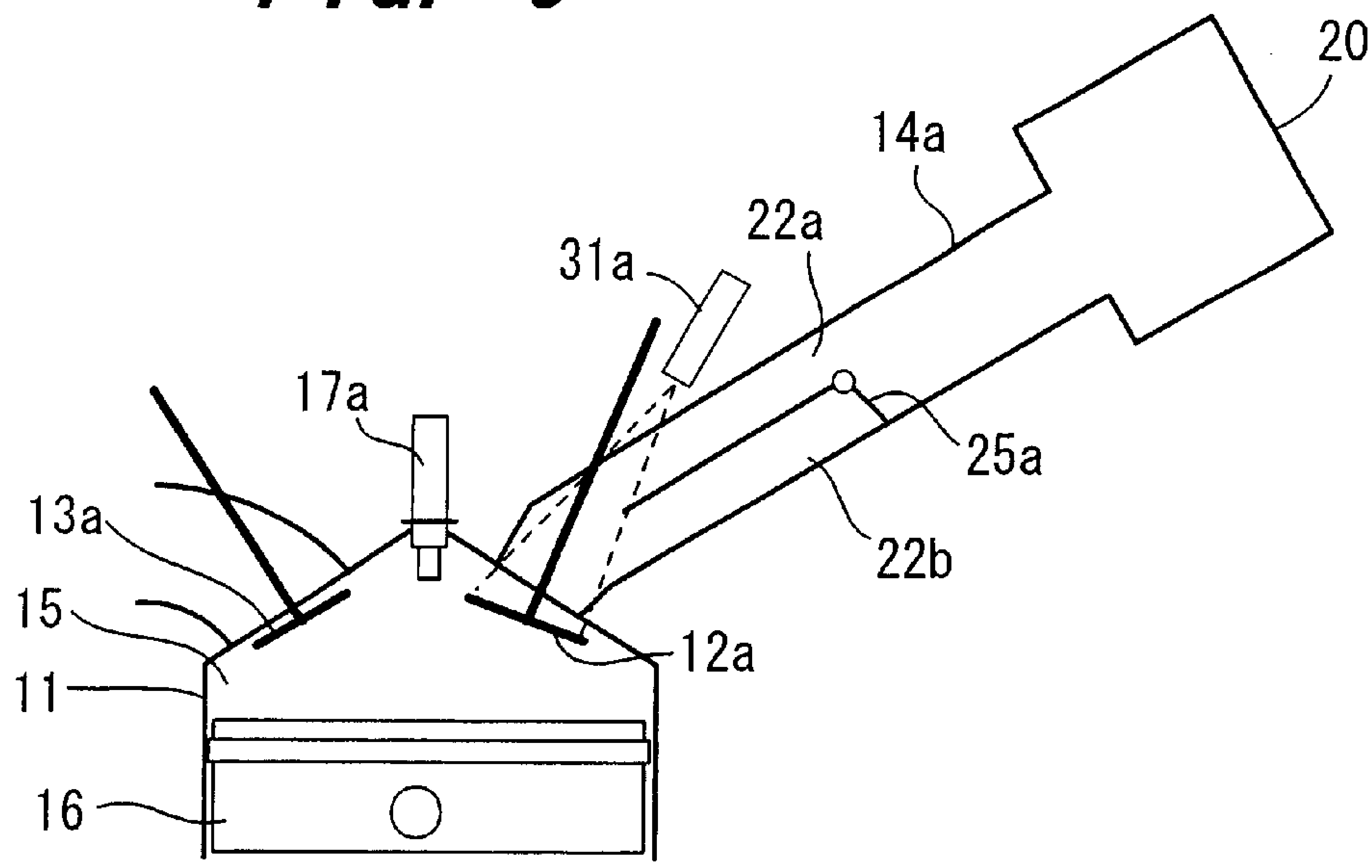

20
14a
22a
31a
17a
25a
13a
22b
15
12a
11
16

FIG. 10

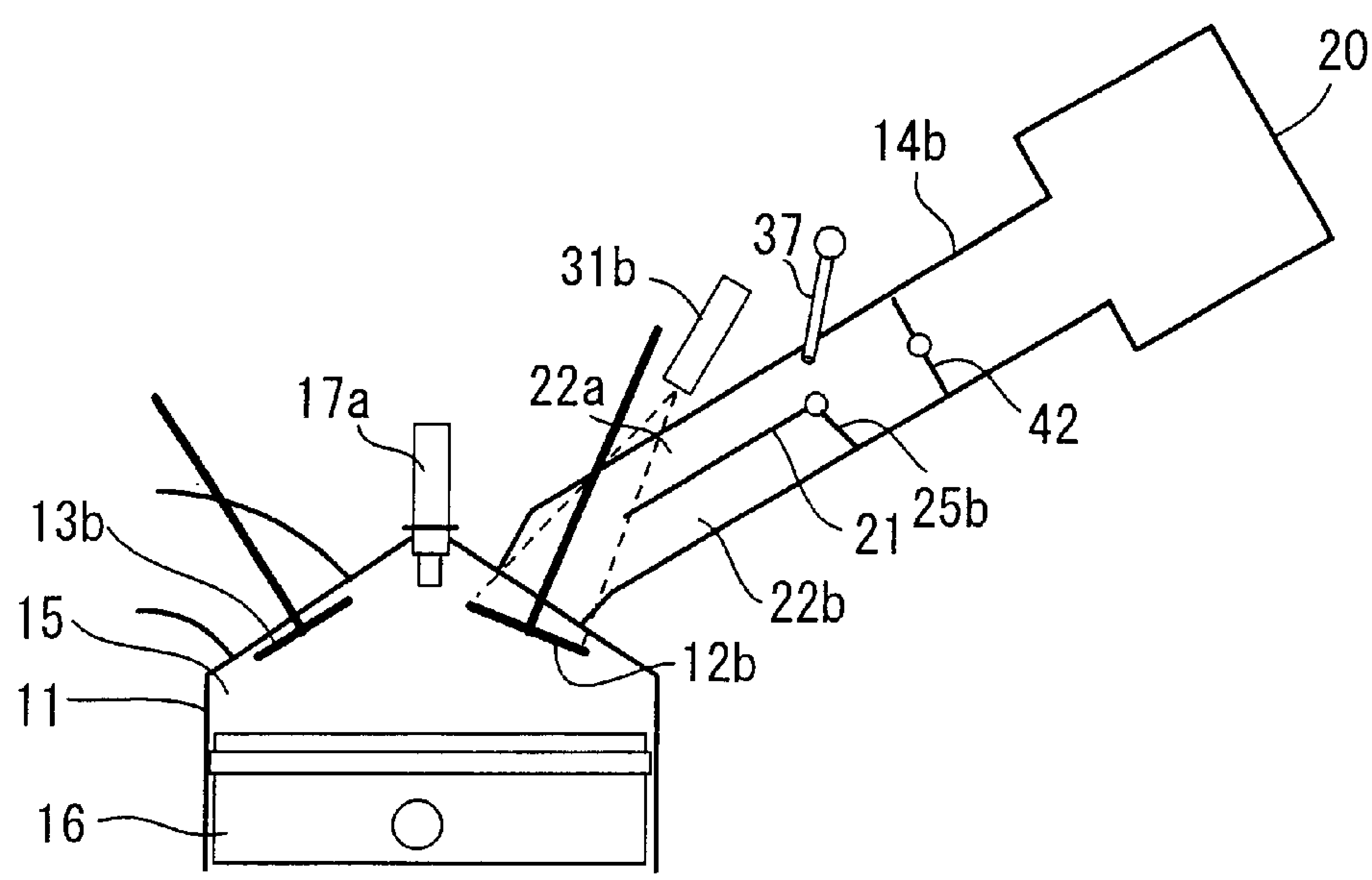
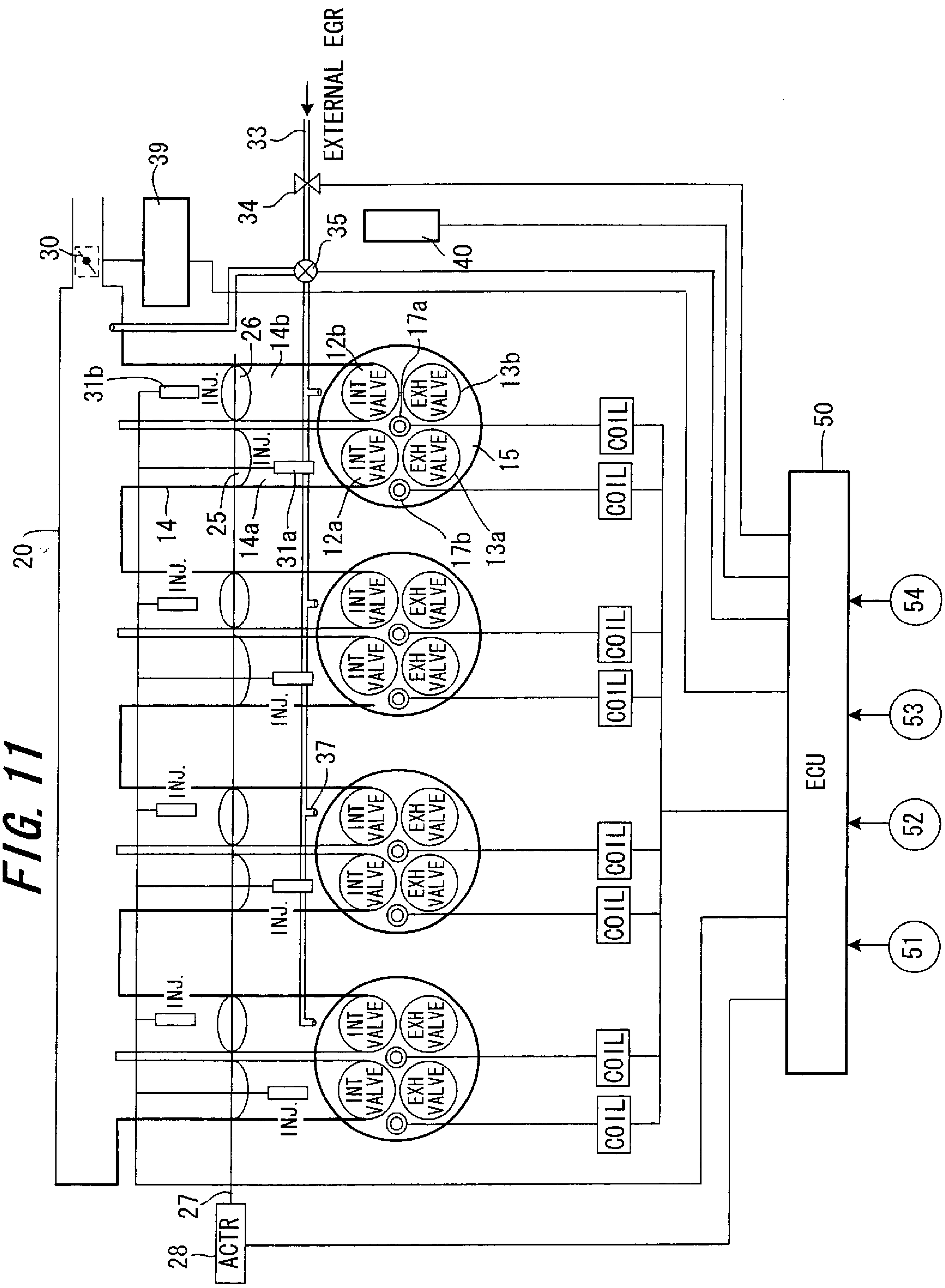

20
14b
37
31b
17a
22a
42
25b
13b
21
22b
15
12b
11
16

FIG. 11



EXTERNAL EGR
33
34
35
39
30
40
20
14
14a
14b
25
26
31a
31b
12a
12b
13a
13b
15
17a
17b
37
INT VALVE
EXH VALVE
INJ.
COIL
ECU
50
51
52
53
54
27
28
ACTR

FIG. 12

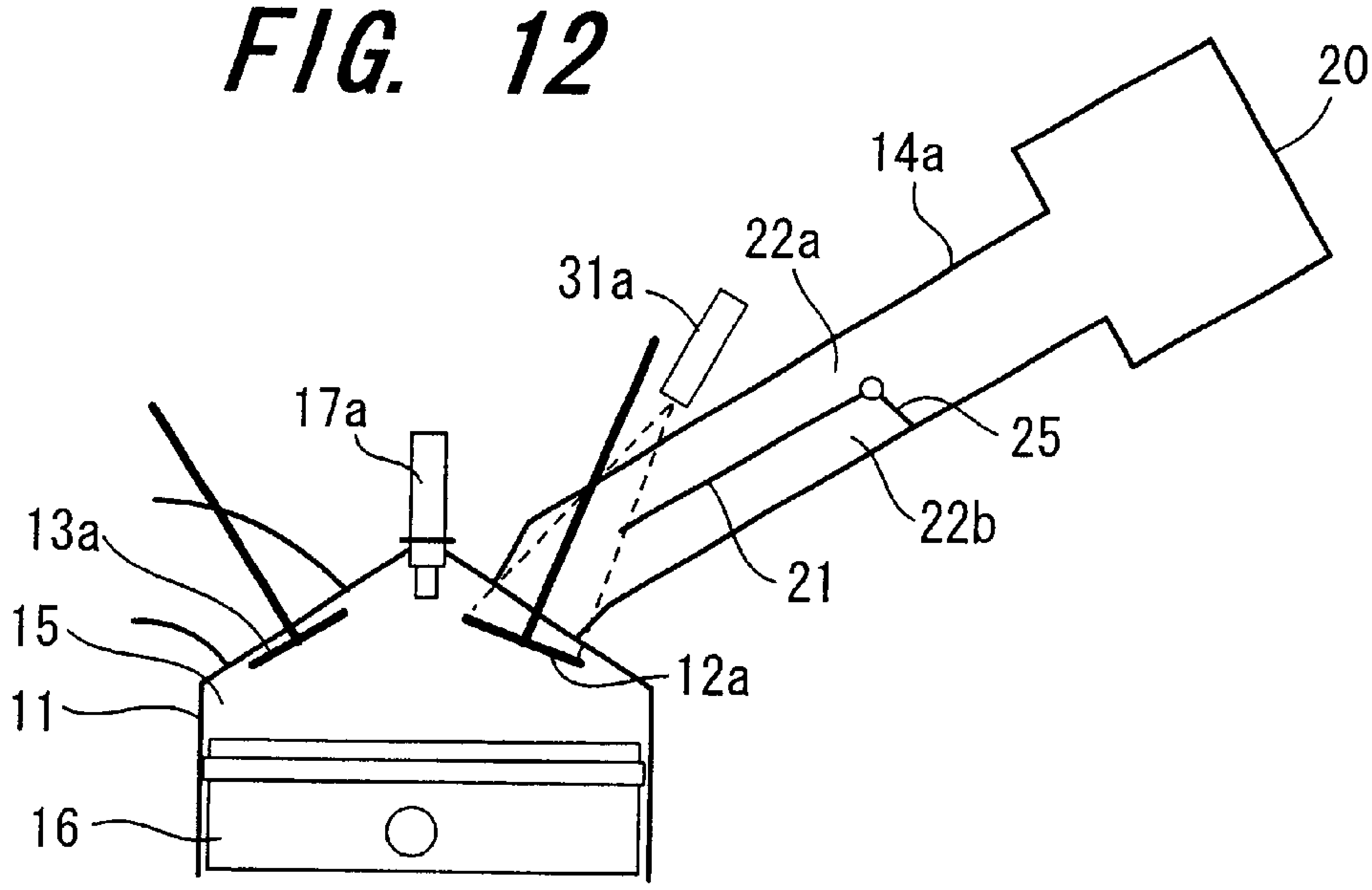

20
14a
22a
31a
17a
25
22b
13a
21
15
12a
11
16

FIG. 13

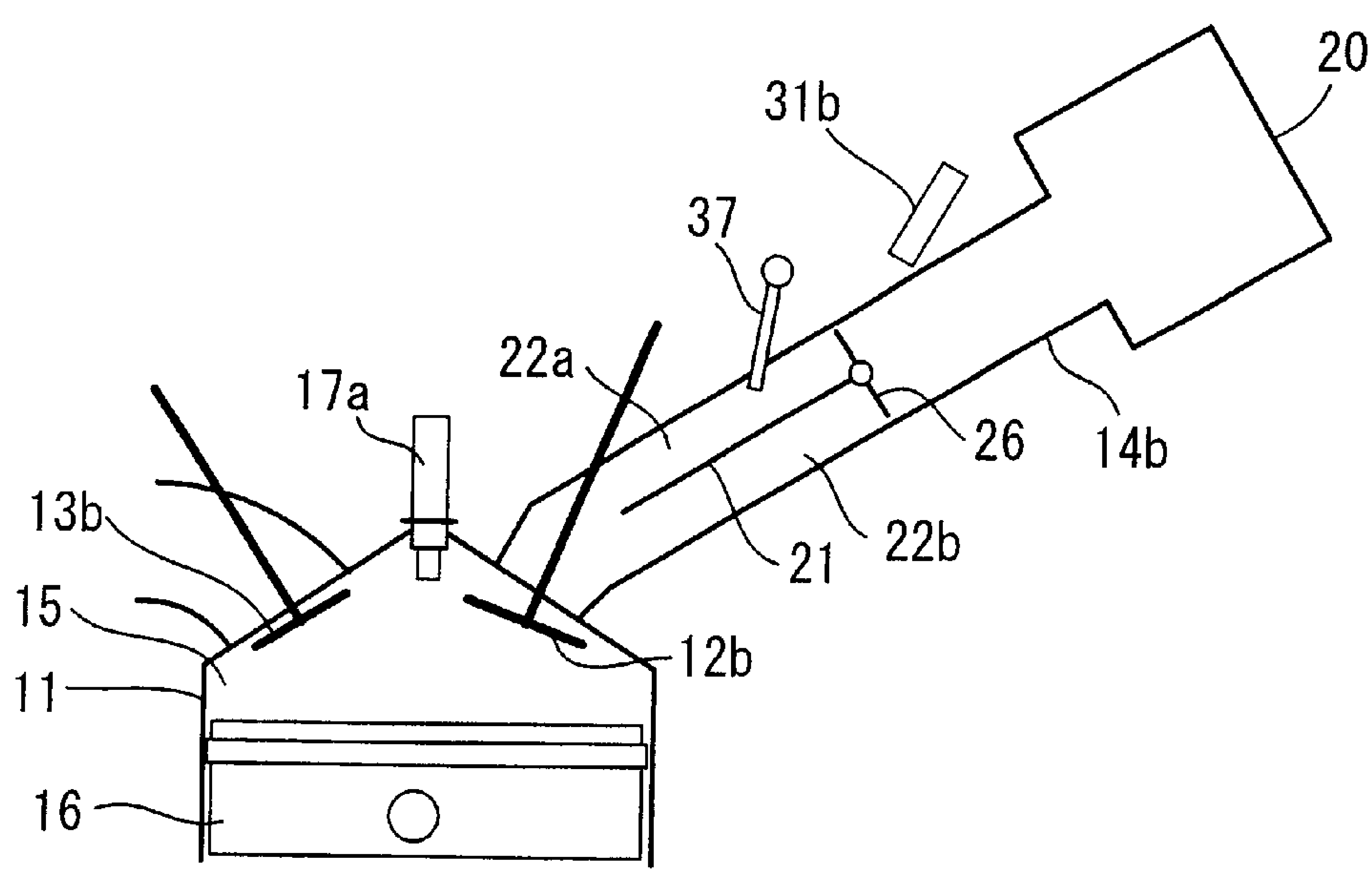

20
31b
37
22a
17a
26
14b
13b
21
22b
15
12b
11
16

FIG. 14

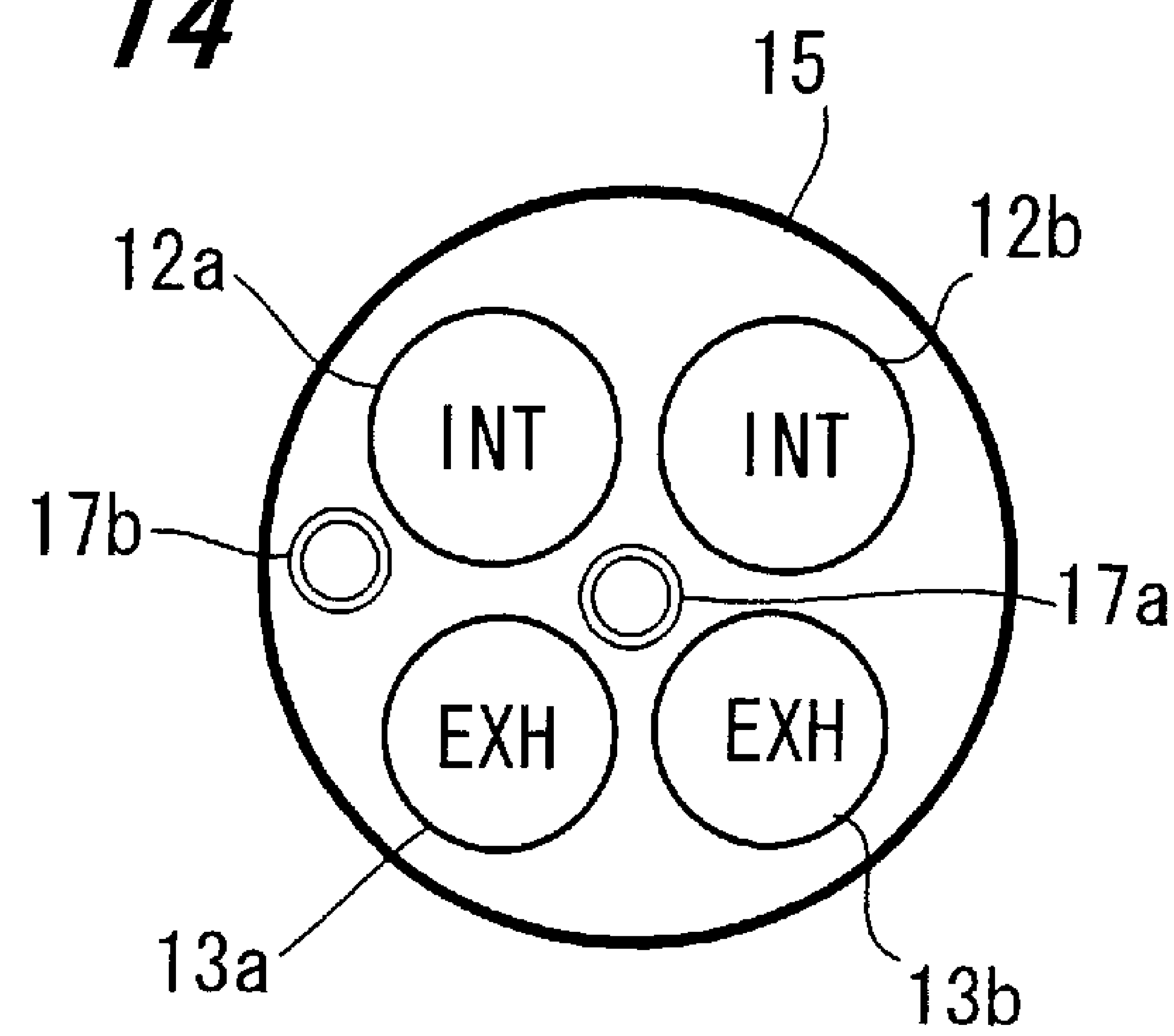


FIG. 15

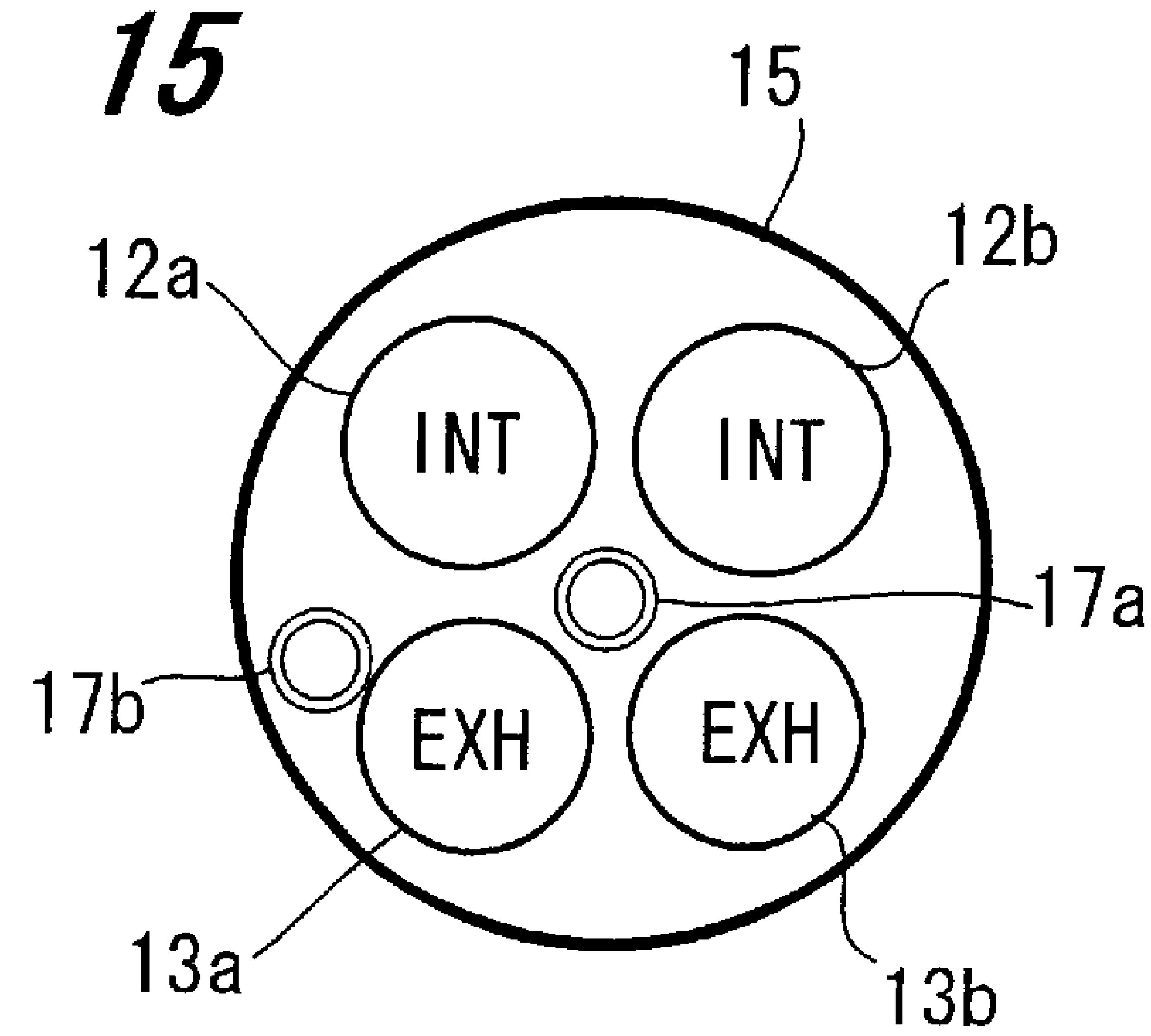


FIG. 16

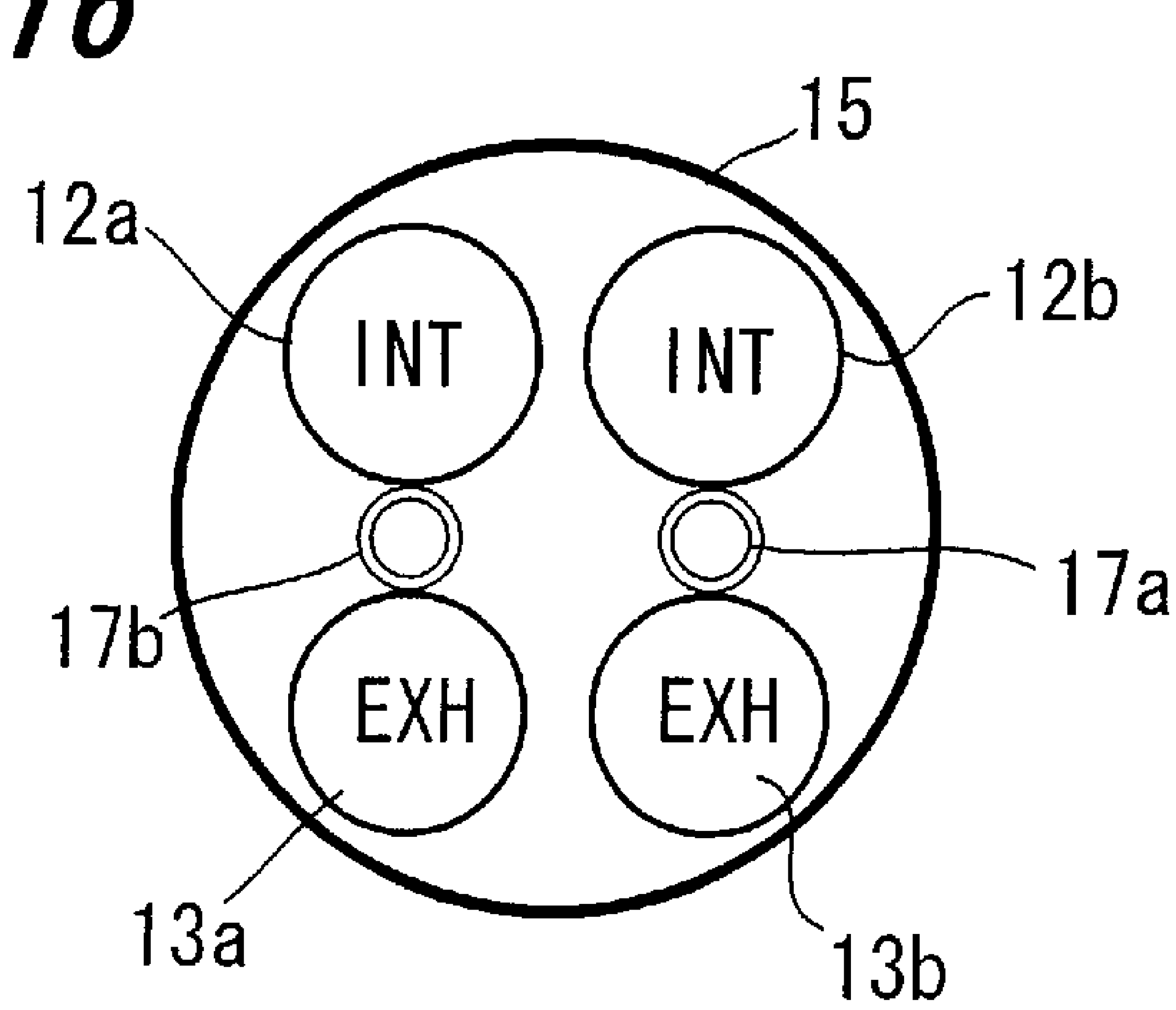


FIG. 17

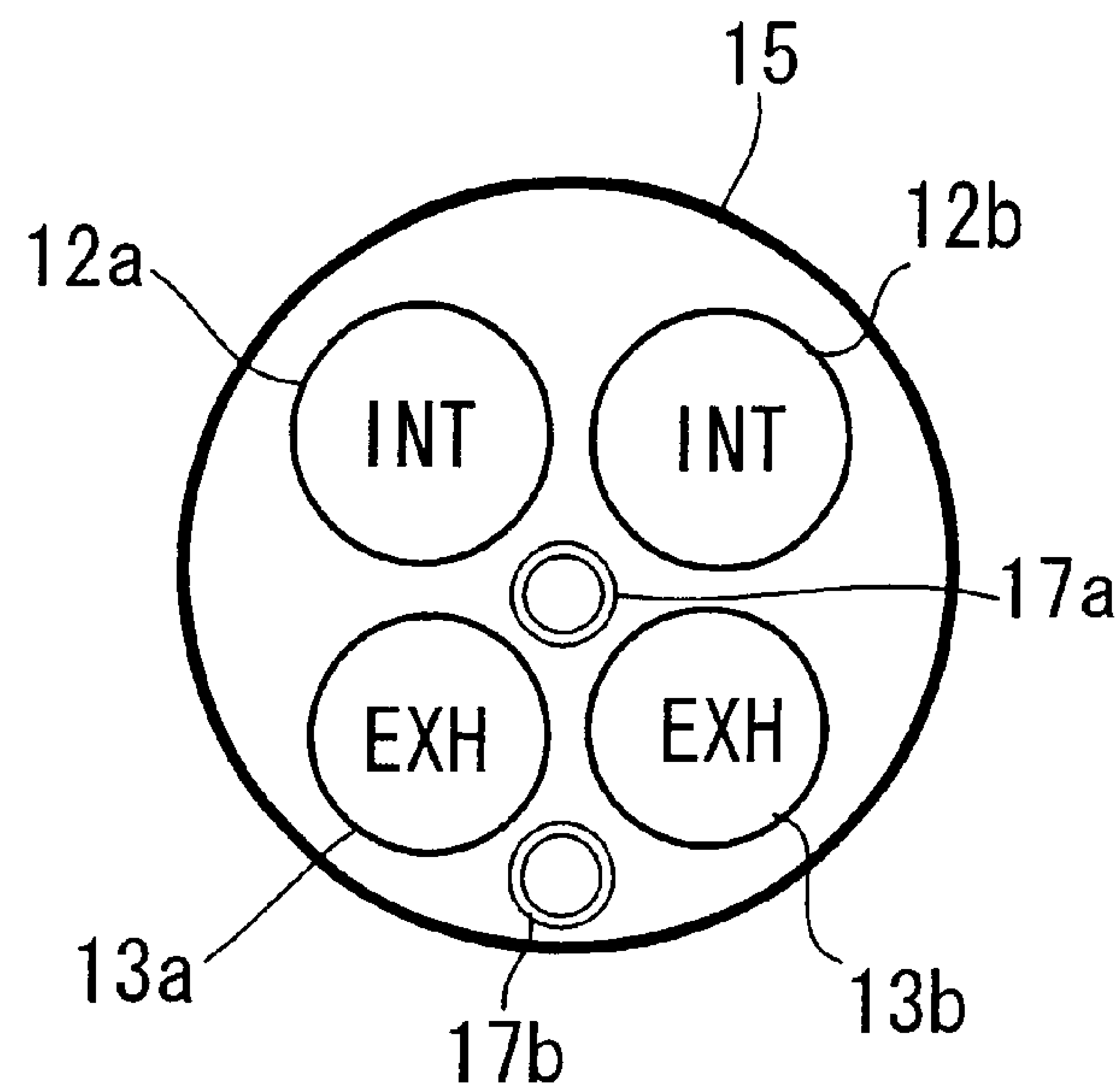
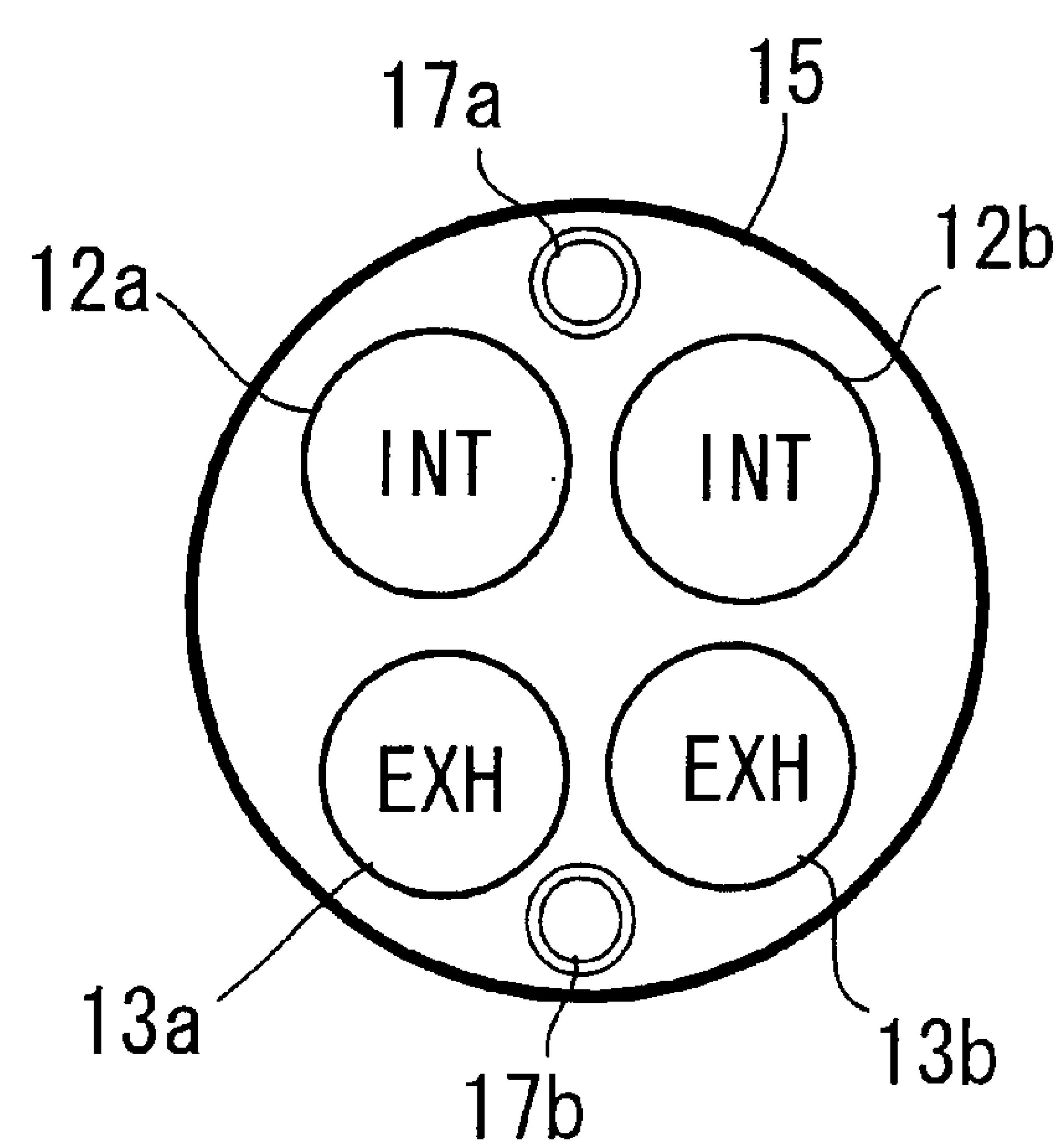

15
12a
12b
INT
INT
17a
EXH
EXH
13a
17b
13b

FIG. 18


17a
15
12a
12b
INT
INT
EXH
EXH
13a
17b
13b

ENGINE CONTROL APPARATUS

TECHNICAL FIELD OF THE INVENTION

This invention relates to a control apparatus for stabilizing combustion in an engine.

TECHNICAL BACKGROUND OF THE INVENTION

A device for controlling gas flow in a combustion chamber to improve the combustion characteristic of an engine by providing two independent intake ports, a primary port and a secondary port, in a cylinder, disposing a control valve upstream of the secondary port, and opening and closing the control valve in accordance with the operating conditions, is proposed in Japanese Patent Kokai Publication JP5-163950A.

SUMMARY OF THE INVENTION

In this case, during engine low load condition the control valve is closed and air-fuel mixture is supplied only from the primary port such that a swirl is generated inside the combustion chamber and the combustion characteristic is improved. However, improvement of the combustion characteristic is limited by the swirl alone. Moreover, improvements in early catalytic activation when the engine was cold proved difficult.

An object of this invention is to provide an engine control apparatus which is designed to be capable of enhancing early catalytic activation and ensuring stable combustion even with a lean air-fuel mixture by stratifying the air-fuel mixture layer in the combustion chamber when the engine is cold condition.

In order to achieve above the object, this invention provides a control apparatus for an internal combustion engine having an intake valve and an exhaust valve, comprising at least two independent intake ports connected to an engine combustion chamber; a first fuel injection valve provided in the first intake port; and a second intake control valve for opening and closing the second intake port on the upstream side of the intake valve, wherein, when the engine is cold condition, the second intake control valve is slightly opened, fuel is injected from the first fuel injection valve, air-fuel mixture from the first intake port and a smaller amount of fresh air from the second intake port are led into the combustion chamber, and thus the overall air-fuel ratio becomes slightly lean.

The details as well as other features and advantages of the invention are set forth in the remainder of the specification and are shown in the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic diagram of a first embodiment of this invention.

FIG. **2** is a cross section of a primary intake port.

FIG. **3** is a cross section of a secondary intake port.

FIG. **4** is a timing chart showing an operation.

FIG. **5** is an illustrative view of the operational characteristic of a spark plug.

FIG. **6** is a timing chart showing a different operation of the first embodiment.

FIG. **7** is a timing chart showing a different operation.

FIG. **8** is a schematic diagram of a second embodiment of this invention.

FIG. **9** is a cross section of a primary intake port.

FIG. **10** is a cross section of a secondary intake port.

FIG. **11** is a schematic diagram of a third embodiment of this invention.

FIG. **12** is a cross section of a primary intake port.

FIG. **13** is a cross section of a secondary intake port.

FIG. **14** is a schematic plan view of a combustion chamber showing an aspect of spark plug disposal.

FIG. **15** is a schematic plan view of a combustion chamber showing different aspect of spark plug disposal.

FIG. **16** is a schematic plan view of a combustion chamber showing further different aspect of spark plug disposal.

FIG. **17** is a schematic plan view of a combustion chamber showing further different aspect of spark plug disposal.

FIG. **18** is a schematic plan view of a combustion chamber showing further different aspect of spark plug disposal.

PREFERRED EMBODIMENTS OF THE INVENTION

Embodiments of this invention will be described below on the basis of the drawings.

In a first embodiment shown in FIGS. **1** through **3**, an engine main body comprises four cylinders **11**, each cylinder **11** having two intake valves **12***a*, **12***b* and two exhaust valves **13***a*, **13***b*. An intake port is constituted by a mutually independent primary port (first intake port) **14***a* and secondary port (second intake port) **14***b* which serve as an intake port **14**. The intake valves **12***a*, **12***b* open and close an opening portion from the primary port **14***a* and secondary port **14***b* to a combustion chamber **15** inside the cylinder.

A piston **16** is disposed within the cylinder, and two spark plugs **17***a*, **17***b* are attached to the combustion chamber **15** in a position above the piston **16**. As regards the disposal of the spark plugs **17***a*, **17***b*, one of the spark plugs **17***a* is disposed near the center of the combustion chamber, and the other spark plug **17***b* is disposed on the periphery of the combustion chamber further toward the outside than the intake valve **12***a* and exhaust valve **13***a* on the primary side and on a cylinder series center line O**2** which connects the center of each cylinder.

When air-fuel mixture inside the combustion chamber is ignited by the spark plugs **17***a*, **17***b*, the resultant combustion energy causes the piston **16** to perform a reciprocating motion, whereupon a crankshaft, not shown, rotates and engine output is produced.

A catalyst, not shown, is provided in an exhaust passage connected to an exhaust port **18** for purifying the exhaust gas which is discharged from the combustion chamber **15** when the exhaust valves **13***a*, **13***b* are opened.

The primary port **14***a* and secondary port **14***b* are formed in parallel to each other on a plane perpendicular to a cylinder axis O**1** which is in alignment with the cylinder shaft, and are disposed equally on the two sides of an intersecting line O**3** which passes through this cylinder axis O**1** and intersects the above-mentioned cylinder series center line O**2** at a right angle.

Hence the primary port **14***a* and secondary port **14***b* are disposed substantially symmetrically around the intersecting line O**3** which passes through the center of the cylinder, thus constituting a so-called straight port. As a result, when intake air is introduced into the combustion chamber from the primary port **14***a* and secondary port **14***b* respectively, the majority thereof forms mutually independent and equal tumble flows in the combustion chamber. Further, when intake air is introduced from only one of the intake ports, the primary port **14***a* for example, a swirl flow component is produced along the inner periphery of the cylinder as well as a tumble flow component, thereby producing a composite gas flow.

The upstream side of the intake port **14** is connected to a collector **20**, and the primary port **14***a* and secondary port **14***b* form mutually independent ports up to the collector **20**.

As is understood from FIGS. **2** and **3**, the interior of the intake port **14** (primary port and secondary port), which is formed at an incline with respect to the cylinder axis O**1**, is partitioned into upper and lower channels **22***a* and **22***b* by a partition wall **21** disposed substantially in the direction of the port axis. The partition wall **21** is not provided along the entire length of the intake port **14**, but starts from a position slightly upstream of the middle position of the intake port **14**, the tip thereof extending to the combustion chamber side to an extent that the tip does not interfere with the intake valves **12***a*, **12***b*, and/or to a position at which fuel spray from fuel injection valves **31***a*, **31***b* does not directly impinge thereon. In effect, this partition wall **21** should be long enough to provide strong directivity, or in other words inertial force, to the flow of intake air flowing through the upper and lower channels **22***a*, **22***b* so that the intake air flow in the combustion chamber produces a gas flow which is strongly influenced by the height, directional properties of the channels **22***a*, **22***b*.

As illustrated in FIG. **2**, the primary port **14***a* is provided with a half-closing type first intake control valve (to be referred to as "half-closing valve" below) **25** which is positioned at the inlet part of the channels **22***a*, **22***b* partitioned by the partition wall **21** and which is capable of opening and closing only the downstream channel **22***b*.

Further, as illustrated in FIG. **3**, the secondary port **14***b* is provided with a fully-closing type second intake control valve (to be referred to as "fully-closing valve" below) **26** which is similarly positioned at the inlet part of the channels **22***a*, **22***b* and which is capable of simultaneously opening and closing both the upper and lower channels **22***a* and **22***b*. As illustrated in FIG. **1**, the half-closing valve **25** and fully-closing valve **26** are both attached to the same rotary shaft **27** which is disposed so as to pass through each intake port **14** in the direction of the cylinder series. The rotary shaft **27** is rotated by a rotary actuator **28** coupled to the end portion thereof to form a constitution in which the half-closing valve **25** and the fully-closing valve **26** open and close by rotating synchronously with each other.

The primary port **14***a* and secondary port **14***b* are also respectively provided with fuel injection valves **31***a*, **31***b* which are positioned downstream of the half-closing valve **25** and fully-closing valve **26**. The position and direction of these fuel injection valves **31***a* and **31***b* are set such that fuel can be injected toward the interior of the combustion chamber from the rear face of the intake valves **12***a* and **12***b*, and preferably such that fuel can be injected uniformly through the upper and lower channels **22***a* and **22***b* without collision the partition wall **21**.

An exhaust gas recirculation passage **33** is provided for recirculating a part of the exhaust gas through the intake air. This exhaust gas recirculation passage **33** is provided with an exhaust gas recirculation control valve **34** for controlling the amount of recirculated exhaust gas and a directional control valve **35** on the downstream side thereof. The exhaust gas recirculation passage **33** is connected to the aforementioned collector **20** downstream of a throttle valve **30**, and a branch passage **36** branches off at the directional control valve **35**. This branch passage **36** is connected via a recirculation port **37** to only the secondary port **14***b* of the intake port **14** in each cylinder on the downstream side of each fully-closing valve **26**.

The directional control valve **35** switches the flow direction of the recirculating exhaust gas flowing through the exhaust gas recirculation passage **33**, and hence when the directional control valve **35** is switched, recirculating exhaust gas is introduced into the collector **20** or introduced into the secondary port **14***b* from the branch passage **36**. In this case, when recirculating exhaust gas is introduced into the collector **20**, the gas flows therefrom into the primary port **14***a* and secondary port **14***b* at an equal concentration.

The throttle valve **30** controls the amount of air intake into the engine and is driven by a throttle actuator **39**.

A valve timing variable control mechanism **40** is also provided for variably controlling the operation timing of the intake valves **12***a*, **12***b* and the exhaust valves **13***a*, **13***b*. Thus the open/close timing of the intake valves **12***a*, **12***b* and exhaust valves **13***a*, **13***b* can be uniformly altered in accordance with the operating conditions, and one of the exhaust valves **13***a*, **13***b* can be opened or closed independently while the other remains closed. This valve timing variable control mechanism **40** is also capable of variably controlling the open/close timing of the intake valves **12***a*, **12***b* and exhaust valves **13***a*, **13***b* independently of one another.

A controller (control means) **50** is provided for controlling the respective operations of the rotary actuator **28**, throttle actuator **39**, valve timing variable control mechanism **40**, exhaust gas recirculation control valve **34**, and directional control valve **35** in accordance with the operating conditions, as will be described below. The controller **50** also controls the fuel injection amount from the fuel injection valves **31***a*, **31***b*, and also controls the ignition timing of each of the spark plugs **17***a*, **17***b*.

For this purpose, signals representing the operating conditions such as a rotation speed signal from an engine rotation speed sensor **51**, an intake air amount signal from an intake air amount sensor **52**, an accelerator opening signal from an accelerator opening sensor **53**, and a water temperature signal from an engine water temperature sensor **54** are input into the controller **50**, on the basis of which the operations of each of these areas are controlled as will be described in detail below.

These operations will be described in detail with reference to the timing charts illustrated in FIGS. **4**, **6**, and **7**.

These timing charts illustrate, in sequence from the top down in the ordinate direction, engine rotation speed, vehicle speed, engine cooling water temperature, the degree of opening of the intake control valves **25**, **26**, operation of the secondary side fuel injection valve **31***b*, operation of the primary side fuel injection valve **31***a*, ignition timing of the spark plugs **17***a*, **17***b*, the on/off state of exhaust gas recirculation, the introduction position of the recirculated exhaust gas (collector or port), and the valve timing of the intake valves and exhaust valves, and in the abscissa direction illustrate the engine operating conditions, these being, in sequence from left to right, an engine start-up period, a fast idling period, an engine partial period (cold), an engine partial period (stratified combustion low load when hot), an engine partial period (medium/high load when hot), and a full load period.

FIG. **6** mainly differs from FIG. **4** in the respective operations of the intake control valve, secondary side fuel injection valve, spark plugs, and exhaust valves during the fast idling period, and FIG. **7** differs from FIG. **4** in the respective operations of the intake control valve, secondary side fuel injection valve, and exhaust valves during the partial period (cold). All other operations are identical to FIG. **4**.

1. Engine Start-Up Period

As shown in the timing chart in FIG. **4**, during operating conditions in which combustion is difficult to stabilize, such as during engine start-up, the half-closing valve **25** of the primary port **14***a* is closed by the controller **50**, as is the fully-closing valve **26** of the secondary port **14***b*. As a result, when the intake valves **12***a*, **12***b* are opened, intake air flows only into the upper channel **22***a* of the primary port **14***a*. It should be noted that at this time, the exhaust gas recirculation control valve **34** is also fully closed and exhaust gas recirculation is halted. The valve opening timing of the intake valves **12***a*, **12***b* (IVO) is set later than intake top dead center, and the two spark plugs **17***a* and **17***b* ignite simultaneously.

The fuel injection amount is set to an appropriate flow rate for start-up in accordance with the intake air amount and the rotation speed, but fuel is supplied only from the fuel injection valve **31***a* of the primary port **14***a*. Thus the air and fuel in the upper channel **22***a* of the primary port **14***a* mix and flow into the combustion chamber **15**.

At this time, the intake air flow which has entered the upper channel **22***a* from upstream of the half-closing valve **25** is accelerated to reach a sufficiently high speed in the channel **22***a* which has a cross section reduced to half that of a normal intake port, and, since the opening timing of the intake valve **12***a* is later than normal opening timing, fuel atomization and vaporization of the fuel injected into the intake port is sufficiently accelerated.

As is also shown in FIG. **2**, the intake air flow smoothly enters the combustion chamber **15** from the upper channel **22***a* through one of the intake valves **12***a*, and since this intake valve **12***a* is in an offset position from the center of the combustion chamber, a strong gas flow with a swirl flow as a central flow component is produced inside the combustion chamber due to the high-velocity intake air flow provided with sufficient directivity in the upper channel **22***a*.

As a result, fuel and air mixing is further accelerated, and despite being under the engine low temperature condition a sufficiently vaporized air-fuel mixture is ignited by the two spark plugs **17***a*, **17***b* simultaneously. Thus stable ignition is performed and the resultant combustion flame propagates swiftly along the gas flow through the combustion chamber to thereby realize stable combustion during start-up.

By increasing combustion stability during start-up in this manner, the increasing amount of fuel during start-up does not have to be significantly increased, leading to a reduction in fuel consumption.

Further, as shown in FIG. **5**, the ignition timing of the peripheral spark plug **17***b* is made earlier than that of the spark plug **17***a* in the center of the combustion chamber rather than the ignition timing of the two spark plugs **17***a*, **17***b* being simultaneous, and although these ignition timings both advance in accordance with increases in the engine rotation speed, the degree of advance in the ignition timing of the central spark plug **17***a* relative to increases in engine speed is larger. Thus as the engine speed increases, from a certain engine speed boundary the central spark plug **17***a* starts to ignite earlier than the peripheral spark plug **17***b*.

As a result, engine speed variation during start-up which arises from the fuel properties can be suppressed by controlling the phase difference in the ignition timing. Alternatively, by igniting the spark plug **17***a* in the center of the combustion chamber first and igniting the peripheral spark plug **17***b* thereafter when the engine speed drops following the upsurge in engine rotation directly after start-up, the reduction in engine torque is compensated for and thus the drop in engine speed may be suppressed.

In so doing, the amount of increase in start-up fuel may be reduced from conventional engine.

2. Fast Idling Operating Period

During fast idling, similarly to the aforementioned start-up period, the controller **50** closes the half-closing valve **25** and the fully-closing valve **26** of the primary port **14***a* of the intake port **14**, and thus when the intake valves **12***a*, **12***b* open, intake air flows only into the upper channel **22***a* of the primary port **14***a*.

Thus fuel is injected only from the fuel injection valve **31***a* on the primary side and the fuel injection amount is set to a slightly lean air-fuel ratio (in other words, excess air ratio (is set as (=1.1 approximately). Note that the exhaust gas recirculation control valve **34** is fully closed and exhaust gas recirculation is halted.

The opening timing of the intake valves **12***a*, **12***b* (IVO) is set later than intake top dead center, the closing timing thereof (IVC) is set at bottom dead center, the opening timing of the exhaust valves **13***a*, **13***b* (EVO) is set later than bottom dead center, and the closing timing thereof (EVC) is set to be earlier than top dead center.

On the spark plugs **17***a* and **17***b*, the ignition timing of the peripheral spark plug **17***b* is set to be later (retarded) than that of the spark plug **17***a* in the center of the combustion chamber.

As a result, air-fuel mixture flows into the combustion chamber **15** only from the primary port **14***a* to produce a strong swirl. Moreover, due to the acceleration in fuel vaporization caused by the intake valves **12***a*, **12***b* opening later than usual, the combustion speed of the air-fuel mixture in the combustion chamber increases such that a stable combustion characteristic is ensured even with a lean air-fuel ratio. Further, by igniting the air-fuel mixture in the central portion first using the spark plug **17***a* in the center of the combustion chamber and igniting the remaining peripheral air-fuel mixture using the peripheral spark plug **17***b* which ignites later, favorable combustion can be maintained in all regions of the combustion chamber.

Hence during the fast idling period, combustion stability is improved and fuel consumption is further reduced. Further, by making the air-fuel ratio slightly lean, early catalytic activation is attained to thereby improve emissions.

The following control may be performed instead of the control described above.

That is, as is illustrated in the timing chart in FIG. **6**, the fully-closing valve **26** of the secondary port **14***b* is slightly opened such that a small amount of intake air also flows from the secondary port **14***b*, and fuel is also injected from the fuel injection valve **31***b* on the secondary side. At this time, the fuel injection valve **31***a* on the primary side injects an amount of fuel to form a lean air-fuel ratio with the intake air flow from the primary port **14***a*, and the fuel injection valve **31***b* on the secondary side injects an amount of fuel to form a rich air-fuel ratio with the small amount of intake air flow on the secondary side. The respective fuel injection amounts are set so that when both air-fuel mixtures are combined, the average air-fuel ratio is lean.

The lean air-fuel ratio from the primary port **14***a* and the rich air-fuel ratio from the secondary port **14***b* exist in the combustion chamber in a layered state and form a gas flow comprising a swirl component.

In this case, even if a slight unevenness occurs in the fuel density due to irregularities in the air-fuel mixture concentration, comparatively stable combustion is performed, similarly to when the half-closing valve **25** and fully-closing valve **26** are fully closed, by means of the strong gas flow possessing sufficient directivity. In this case a comparatively large amount of unburned HC and the like is contained in the exhaust gas, and that by burning this unburned HC and the like in the exhaust system such that the exhaust system is caused to function as a thermal reactor, high temperature exhaust gas is led to the catalyst, causing the catalyst temperature to rise. In other words, the catalyst is heated as quickly as possible when the engine is in the cold condition. Furthermore, at this time the total air-fuel ratio in the entire combustion chamber is set to slightly lean (for example (=1.1), and thus an oxidation reaction is maintained in the catalyst to thereby accelerate early activation.

Further, by setting both of the spark plugs **17***a*, **17***b* to relative retarded sides, combustion delay can be increased, and by setting the opening timing of the exhaust valves **13***a*, **13***b* (EVO) to an early timing, high temperature exhaust gas containing many unburned components can be discharged to thereby increase the reaction in the catalyst and further accelerate the early activation thereof.

Note that instead of opening the exhaust valves **13***a*, **13***b* at an early timing in this manner, any one of the exhaust valves **13***a* or **13***b* may be left closed. In this case, exhaust gas flows into the exhaust port from only one of the exhaust valves **13***a* or **13***b*, and since the surface area of the port into which the exhaust gas flows may be reduced, decreases in the exhaust gas temperature can be reduced such that the temperature of the catalyst rises and early activation is accelerated.

3. Period of Partial Operation when the Engine is Cold

In FIG. **4**, control is performed in basically the same manner during the period of partial operating when the engine is cold (partial load operating period) as during the fast idling period.

When air-fuel mixture is introduced only from the primary port **14***a*, the combustion characteristic is improved on the basis of a strong swirl in the combustion chamber, whereby the air-fuel ratio in the cold state can be made lean and fuel consumption can be reduced.

On the other hand, when the half-closing valve **25** and fully-closing valve **26** are slightly opened and fuel is injected only from the fuel injection valve **31***a*, lean air-fuel mixture is introduced from the upper and lower channels **22***a*, **22***b* of the primary port **14***a*, whereas only a small amount of fresh air is introduced from the secondary port **14***b*. As a result, a stratified state with a tumble flow as the main flow component thereof is formed in the combustion chamber from an air-fuel mixture layer from the primary side and a fresh air layer from the secondary side. By setting the total air-fuel ratio at this time to be lean, good stratified combustion of the lean air-fuel mixture is performed.

As a result, the combustion characteristic during the cold period can be improved, and fuel consumption can be reduced. The determination of a cold condition is performed on the basis of the cooling water temperature detected by the engine cooling water temperature sensor **54**, and this control may be switched on and off in accordance with the cooling water temperature. It should be noted that at this time, in order to ensure combustion stability in a cold condition, exhaust gas recirculation is halted.

In order to raise the catalyst temperature and achieve early catalytic activation, the opening timing of the exhaust valves **13***a*, **13***b* (EVO) is brought largely advanced and the closing timing thereof (EVC) is slightly delayed, as is shown in the timing chart in FIG. **7**. The opening timing of the intake valves **12***a*, **12***b* (IVO) at this time is also delayed. In so doing, high temperature exhaust gas flows into the exhaust port **18**, thereby raising the temperature of the exhaust system and enabling a rapid temperature increase in the catalyst.

Once the cooling water has risen to a predetermined temperature, processing moves to control when the engine is hot, to be described herein below.

4. Period of Partial Low Load Operating when the Engine is Hot

In the period of low load operating when the engine is hot, exhaust gas recirculation is performed to achieve a reduction in NOx. Also in this low load operating condition, the half-closing valve **25** of the primary port **14***a* and the fully-closing valve **26** of the secondary port **14***b* are closed and also connected to the branch passage side by the directional control valve **36**. The exhaust gas recirculation control valve **34** is opened to a degree in accordance with the amount of intake air.

Thus intake air flows through the upper channel **22***a* in the primary port **14***a* to then flow via the intake valve **12***a* into the combustion chamber **15** maintaining sufficient directivity. Further, recirculated exhaust gas flows into the upper channel **22***a* of the secondary port **14***b* from the recirculation port **37**, and when the intake valve **12***b* opens, recirculated exhaust gas also flows into the combustion chamber **15** maintaining sufficient directivity. In this case, fuel is injected only from the fuel injection valve **31***a* of the primary port **14***a* and the air-fuel ratio is set to a substantially stoichiometric level.

The ignition timing of the spark plugs **17***a* and **17***b* and the open/close timing of the exhaust valves **13***a* and **13***b* are identical to the cold partial period, whereas the opening timing of the intake valves **12***a*, **12***b* (IVO) is advanced and the closing timing thereof (IVC) is set at intake bottom dead center in order to increase the amount of exhaust gas recirculated into the combustion chamber.

Since the primary port **14***a* and secondary **14***b* are parallel to one another and the intake valves **12***a* and **12***b* are arranged in equal positions on each side of the cylinder axis, the greater parts of the intake air and recirculated exhaust gas having strong directivity remain in a mutually parallel state inside the combustion chamber and therefore flow in the direction of the cylinder axis to form a tumble flow. The air-fuel mixture and recirculated exhaust gas in the combustion chamber can therefore be stratified as a tumble flow mainly comprising an air-fuel mixture layer which is then ignited by the spark plugs **17***a*, **17***b* positioned on both sides of the tumble flow, and thus even an air-fuel mixture containing a large amount of recirculated exhaust gas can be burned with stability. As a result, a vast reduction in fuel consumption due to large-scale EGR (exhaust gas recirculation) and suppression of NOx can be achieved with no deterioration in operating performance.

If the amount of exhaust gas recirculation is controlled by the exhaust gas recirculation control valve **34** such that the ratio of intake air from the primary port **14***a* and recirculated exhaust gas from the secondary port **14***b* becomes 1:1, the two flow rates are equalized, or in other words the left and right side flows become symmetrical, whereby gas flow stratification in the combustion chamber is performed most favorably. As a result, the EGR threshold region is extended such that, regardless of the large amounts of exhaust gas recirculation, combustion stability can be maintained and further reductions in fuel consumption and NOx can be realized.

In this low load period, if exhaust gas recirculation is halted and the half-closing valve **25** and fully-closing valve **26** are fully opened such that intake air is introduced from both the primary port **14***a* and secondary port **14***b* and fuel is injected from only one fuel injection valve, for example the fuel injection valve **31***a* on the primary side, an identical uniform tumble flow to that described above can be generated in the combustion chamber. In this case, the flow which enters from the primary side becomes the air-fuel mixture layer and the flow which enters from the secondary side becomes the air layer, and as a result of this stratification, stable stratified lean combustion can be realized even with an extremely lean total air-fuel ratio. In so doing, engine fuel consumption can be reduced and the exhaust emission can be improved.

5. Period of Partial Medium/High Load Operating when Engine is Hot

In the condition of medium/high load operating when the engine is hot, the half-closing valve **25** of the primary port **14***a* and the fully-closing valve **26** of the secondary port **14***b* are altered from an intermediate to a maximum degree of opening in accordance with the load. As the amount of engine intake air increases, intake efficiency deteriorates if the entire required amount is supplied from only the primary port **14***a*, and therefore intake air is introduced from both of the intake ports **14**. When not in a full load condition (full throttle condition), however, the fully-closing valve **26** of the secondary port **14***b* is not opened fully, but rather controlled to a degree of opening in accordance with the amount of engine intake air.

Fuel is supplied from both of the fuel injection valves **31***a* and **31***b*. Further, in order to mix fresh air and recirculated exhaust gas in advance to perform homogeneous exhaust gas recirculation, the directional control valve **35** is switched so that the recirculated exhaust gas is led to the collector **20** rather than the branch passage **36**. As a result, the intake air and recirculated exhaust gas mix in the collector **20** and flow into both the primary port **14***a* and secondary port **14***b* in a uniform concentration.

Intake air flows from the primary port **14***a* and secondary port **14***b*, and since the half-closing valve **25** and filly-closing valve **26** are open to an intermediate extent, the intake air flows not only from the upper channel **22***a* but also from the lower channel **22***b*.

In this case, only the upper channel **22***a* of the primary port **14***a* is fully open, and thus the flow rate through the primary port **14***a* is somewhat larger than the flow rate through the secondary port **14***b*.

Hence, of the intake air entering the combustion chamber **15**, the swirl flow component heading toward the secondary port **14***b* side from the primary port **14***a* side and the respectively formed tumble flow components combine to generate a composite gas flow.

Since the recirculated exhaust gas is mixed with the air-fuel mixture in advance, a substantially uniform concentration is maintained in all regions of the combustion chamber. When the fuel in the air-fuel mixture is atomized and vaporized by the gas flow generated inside the combustion chamber and subjected to two point ignition by the two spark plugs **17***a*, **17***b*, the propagation distance to be traveled by the flame is shortened, and thus stable combustion can be completed within a short time period even when the air-fuel mixture contains recirculated exhaust gas. As a result, simultaneous reductions in NOx and in fuel consumption can be achieved.

It should be noted that in this medium load operating region, by altering the opening of the half-closing valve **25** and fully-closing valve **26** in accordance with the amount of intake air, the gas flow may be controlled to the most appropriate state for improvements in combustion.

As the opening of the half-closing valve **25** and fully-closing valve **26** increases and the engine load nears a higher load than a predetermined value, the exhaust gas recirculation control valve **34** is closed and exhaust gas recirculation is halted. In so doing, engine output is raised and a favorable operating characteristic can be maintained.

The open/close timing of the intake valves **12***a*, **12***b* and exhaust valves **13***a*, **13***b* is preferably controlled to the most appropriate condition for alleviating pumping loss. For example, the opening timing (IVO) and closing timing (IVC) of the intake valves **12***a*, **12***b* are advanced, the opening timing of the exhaust valves **13***a*, **13***b* (EVO) is altered according to the load, and the closing timing thereof (EVC) is delayed.

6.Engine Full Load (Full Throttle) Operating Period

In the high load operating region following warming, the half-closing valve **25** and filly-closing valve **26** are both fully opened such that intake air is led into the combustion chamber **15** uniformly from both the primary port **14***a* and the secondary port **14***b*. Fuel is injected from both of the fuel injection valves **31***a*, **31***b*, and exhaust gas recirculation is halted.

Since both the half-closing valve **25** and fully-closing valve **26** are fully open, resistance in the intake air flowing through the intake port **14** is at a minimum, and thus engine intake efficiency is at its most favorable and high output is manifested in the engine.

It should be noted that in this operating region the temperature is high, depression is low, and the combustion conditions are favorable, and therefore increases in combustion noise and vibration can be suppressed by performing ignition using the central spark plug **17***a* alone.

A second embodiment will be described with reference to FIGS. **8** through **10**.

The second embodiment differs from the first embodiment in the following points. Firstly, half-closing valves **25***a*, **25***b* are provided as intake control valves in only the lower channel **22***b* of both the primary port **14***a* and the secondary port **14***b*. Further, a shut-off valve **42** is provided upstream of the half-closing valve **25***b* in the secondary port **14***b*. The shut-off valves **42** in each of the cylinders are attached to a rotary shaft **43** in phase with one another, and are open/close driven in unison by a rotary actuator **44**. Further, a recirculation port **37** is opened in a position between the shut-off valve **42** and the half-closing valve **25***b* for introducing recirculated exhaust gas into the secondary port **14***b*.

Thus in this embodiment, control in the engine start-up period, fast idling period, and cold period is performed similarly to the first embodiment with the shut-off valve **42** of the secondary port **14***b* fully closed and the half-closing valves **25***a*, **25***b* closed. In the low load period when the engine is hot, exhaust gas recirculation is performed only from the secondary port **14***b* so that stratification is achieved in the combustion chamber, and thus NOx can be reduced without harming the operating performance.

Further, in the medium load period, by fully opening the shut-off valve **42** and performing exhaust gas recirculation from the collector side while controlling the opening of the half-closing valves **25***a*, **25***b* in accordance with the engine load, the gas flow into the combustion chamber can be controlled appropriately to attain a reduction in NOx and an improvement in the combustion characteristic. High engine output can be generated at the engine full throttle operation by similarly opening the shut-off valve **42** and half-closing valves **25***a*, **25***b* fully with exhaust gas recirculation halted.

Also during the engine fast idling period or the cold period, the shut-off valve **42** is slightly opened and the half-closing valves **25***a*, **25***b* are slightly opened such that intake air also flows from the secondary port **14***b*, although in a smaller amount than from the primary port **14***a*, and fuel injection is also performed from the secondary side in addition to the primary side. At this time, the air-fuel ratio on the low flow rate secondary side is controlled to become rich. Then, by setting average air-fuel ratio both of the primary side and the secondary side to become lean, comparatively stable combustion is maintained by a swirl even when irregularities arise in the air-fuel mixture concentration distribution throughout the combustion chamber, and furthermore, unburned components in the exhaust gas which is discharged from the combustion chamber **15** during the exhaust stroke may be increased to effect early catalytic activation. In this case, during the low load condition when the engine is cold, a swirl can be generated in the combustion chamber to enable combustion with a lean air-fuel mixture by halting fuel injection on the secondary side and causing only a small amount of intake air to flow from the secondary port **14***b*.

Also in this embodiment, the shut-off valve **42** is provided on the upstream of the secondary port **14***b* and identical half-closing valves **25***a*, **25***b* are arranged in the primary port **14***a* and secondary port **14***b* so as to be opened and closed in phase. Thus during the low load period when the engine is hot, for example, the flow of intake air on the primary side and secondary side can be controlled in exactly the same manner, that is the distribution of intake air in the upper channel **22***a* and lower channel **22***b* can be matched, when the shut-off valve **42** is left fully open and the half-closing valves **25***a*, **25***b* are opened and closed in synchronization with each other. As a result, stratification in the combustion chamber of the tumble flow on the primary side and secondary side becomes easy, and when fuel is injected in this state from either of the fuel injection valves **31***a* or **31***b*, an air-fuel mixture layer and an air layer can be formed. As a result of this stratification in the combustion chamber, stable combustion is possible even with a lean air-fuel mixture.

In the case described above, in which exhaust gas is recirculated into the intake port **14** through the recirculation port **37**, the recirculation port **37** can be disposed in an upstream position removed from the fuel injection valve **31***b*, thus preventing recirculated exhaust gas from clogging the fuel injection valve **31***b*.

A third embodiment will be described with reference to FIGS. **11** through **13**.

This third embodiment differs from the first embodiment in that the fuel injection valve **31***b* of the secondary port **14***b* is disposed on the upstream side rather than the downstream side of the fully-closing valve **26**.

In so doing, the fuel injection valve **31***b* is not exposed to high temperature exhaust gas even when exhaust gas is recirculated to the intake port **14** through the recirculation port **37**, and thus the fuel injection valve **31***b* can be protected.

The control operations in this case are basically identical to those of the first embodiment. However, if fuel is injected from the fuel injection valve upstream of the fully-closing valve **26** during the engine fast idling period, when a small amount of intake air is released from the secondary port **14***b* in order to accelerate catalytic activation, a large amount of this fuel may become adhered to the fully-closing valve **26**, and it is therefore possible to release only intake air without injecting fuel into the secondary side.

In the aforementioned first through third embodiments, the half-closing valves **25**, **25***a*, **25***b* are all designed to open and close the lower channel **22***b* of the intake port **14**, but these valves may be disposed so as to open and close the upper channel **22***a*. It should be noted, however, that when large-scale exhaust gas recirculation is performed during the low load operating period when the engine is hot, the lower channel **22***b* of the intake port **14** should preferably be closed, as in the first through third embodiments, in order to ensure combustion stability. In so doing, the air-fuel mixture layer from the primary side and the recirculated exhaust gas layer from the secondary side maintain a strong tumble flow inside the combustion chamber, by means of which favorable stratified combustion is performed.

The primary port **14***a* and secondary port **14***b* are divided internally into the upper and lower channels **22***a*, **22***b* by the partition wall **21**, but the secondary port **14***b* may be formed as a single channel instead of being partitioned by the partition wall **21**.

Other embodiments regarding the disposal of the spark plugs will now be described.

The position of the central spark plug **17***a* in FIG. **14** is identical to FIG. **1**, but the position of the peripheral spark plug **17***b* has been moved slightly closer to the intake valve **12***a*. In FIG. **15**, the position of the central spark plug **17***a* does not change, but the peripheral spark plug **17***b* has been moved slightly closer to the exhaust valve **13***a* side. A favorable combustion characteristic can be ensured in the same manner as FIG. **1** both when exhaust gas recirculation is performed and exhaust gas recirculation is halted.

In FIG. **16**, the spark plugs **17***a* and **17***b* are disposed between the opposing intake valve **12***a* and exhaust valve **13***a*, and the opposing intake valve **12***b* and exhaust valve **13***b*. In this case, during exhaust gas recirculation cessation and homogeneous exhaust gas recirculation, simultaneous ignition is performed by the two spark plugs **17***a*, **17***b*, whereas during stratified exhaust gas recirculation, ignition is mainly performed on the primary side. As a result, an ignition action in both of the required ignition positions can be performed, enabling a favorable combustion characteristic in both positions.

In FIG. **17**, the position of the central spark plug **17***a* is identical to that in FIG. **1**, but the peripheral spark plug **17***b* is disposed between and on the outside of the two exhaust valves **13***a*, **13***b*. During exhaust gas recirculation cessation and homogeneous exhaust gas recirculation, ignition is mainly performed by the central spark plug **17***a*, whereas during stratified exhaust gas recirculation, ignition is performed by the two spark plugs **17***a* and **17***b*. As a result, stable combustion is achieved. In this case, by disposing the peripheral spark plug **17***b* on the outside of the exhaust valves **13***a*, **13***b*, disposal of a cooling water passage on the periphery of the spark plug in the cylinder head is facilitated.

In FIG. **18**, the position of the spark plug **17***b* is identical to that in FIG. **17**, but the spark plug **17***a* is disposed between and on the outside of the two intake valves **12***a*, **12***b*. In so doing, the periphery of the spark plug **17***a* can also be cooled favorably.

It should be noted that in each of these embodiments, the spark plugs **17***a*, **17***b* may be disposed offset from, rather than parallel to, the cylinder axis in order to avoid interference with the intake valves **12***a*, **12***b* or the exhaust valves **13***a*, **13***b* which are disposed in the cylinder head. The entire contents of Japanese Patent Application P2002-117080 (filed Apr. 19, 2002) is incorporated herein by reference.

It goes without saying that this invention is not limited to the aforementioned embodiments and may be subjected to various modifications within the scope of the technical ideas of the invention.

What is claimed is:

1. A control apparatus for an internal combustion engine having an intake valve and an exhaust valve, comprising:

at least two independent intake ports connected to an engine combustion chamber;

a first fuel injection valve provided in the first intake port; and

a second intake control valve for opening and closing the second intake port on the upstream side of the intake valve,

wherein, when the engine is cold condition, the second intake control valve is slightly opened, fuel is injected from the first fuel injection valve, air-fuel mixture from the first intake port and a smaller amount of fresh air from the second intake port are led into the combustion chamber, and thus the overall air-fuel ratio becomes slightly lean.

2. The engine control apparatus as defined in claim **1**, further comprising a first intake control valve for opening and closing a lower side channel from among an upper and lower channel defined by a partition wall which extends in an axial direction through the interior of the first intake port,

wherein, when the engine is cold condition, the lower side channel is also slightly opened by the first intake control valve.

3. The engine control apparatus as defined in claim **1**, further comprising a second fuel injection valve which is positioned in the second intake port on the downstream side of the second intake control valve,

wherein, when the engine is cold condition, fuel is injected from both the first and second fuel injection valves such that the air-fuel mixture in the second intake port becomes leaner than the air-fuel mixture in the first intake port and the overall air-fuel ratio which is led into the combustion chamber becomes slightly lean.

4. The engine control apparatus as defined in claim **3**, further comprising a first intake control valve for opening and closing a lower side channel from among an upper and lower channel defined by a partition wall which extends in an axial direction through the interior of the first intake port,

wherein, when the engine is cold condition, the lower side channel is also slightly opened by the first intake control valve.

5. The engine control apparatus as defined in claim **1**, further comprising an upper and lower channels which are defined by a partition wall extended in an axial direction through the interior of the second intake port, wherein the second intake control valve opens and closes the upper and lower channels, and

when the engine is cold condition, the lower side channel of the second intake port is slightly opened.

6. The engine control apparatus as defined in claim **5**, wherein the first intake control valve of the first intake port and the second intake control valve of the second intake port are mounted on the same rotary shaft and rotate in synchronization with each other.

7. The engine control apparatus as defined in claim **1**, further comprising two spark plugs in the combustion chamber,

wherein the two spark plugs are ignited simultaneously or only one of the spark plugs is ignited depending on the operating conditions.

8. The engine control apparatus as defined in claim **7**, wherein one of the spark plugs is disposed in the center of the combustion chamber and the other is disposed on the periphery of the combustion chamber.

9. The engine control apparatus as defined in claim **7**, wherein one of the spark plugs is disposed in the center of the combustion chamber and the other is disposed on the periphery of the combustion chamber in a position outside of the exhaust valve.

10. The engine control apparatus as defined in claim **7**, wherein one of the spark plugs is disposed in the center of the combustion chamber, and the other is disposed on the periphery of the combustion chamber in a position between and on the outside of the intake valve and exhaust valve on the first intake port side.

11. The engine control apparatus as defined in claim **7**, wherein one of the spark plugs is disposed in the center of the combustion chamber, and the other is disposed on the periphery of the combustion chamber in a position between and on the outside of the two exhaust valves.

12. The engine control apparatus as defined in claim **7**, wherein both of the spark plugs are disposed on the periphery of the combustion chamber, one on the outside of the intake valve and the other on the outside of the exhaust valve.

13. The engine control apparatus as defined in claim **7**, wherein the spark plugs are respectively disposed between each intake valve and exhaust valve pair and on the central side of the combustion chamber.

14. The engine control apparatus as defined in claim **8**, wherein, when the engine is cold condition, the spark plug on the periphery of the combustion chamber is ignited latter than the spark plug in the center of the combustion chamber.

15. The engine control apparatus as defined in claim **1**, further comprising a variable control mechanism for variably and independently controlling the opening and closing timing of the exhaust valves,

wherein, when the engine is cold condition, one of the exhaust valves is maintained in a closed state.

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